AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 42 No. 549. JANUARY, 1952 PRICE : 3s. 6d.

SO ADAPTABLE — **it solves every factory lighting problem**

SO UP-TO-DATE — **it is ideal for modern industry**

Efficient Osram fluorescent lighting such as this plays a most important part in improving working conditions and raising output. Osram fluorescent lamps and G.E.C. fittings are available for all industries.

G.E.C. illuminating engineers will gladly check your existing lighting. Expert advice is freely available for new installations.

If you already have fluorescent lighting, your old lamps may not be giving full value for current consumed. Replace them with OSRAM fluorescent lamps.

Osram

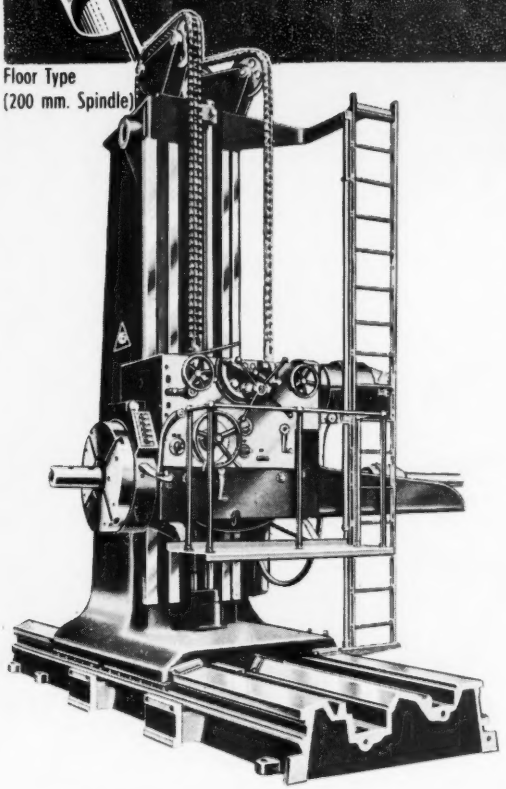
FLUORESCENT LIGHTING with G.E.C. fittings

THE GENERAL ELECTRIC CO. LTD., MAGNET HOUSE, KINGSWAY, LONDON, W.C.2

Wickman for COLLET & ENGELHARD

HORIZONTAL BORING & MILLING MACHINES

Floor Type
(200 mm. Spindle)



TRAVELLING COLUMN (Floor type)

Machines with the boring head mounted on an upright which travels on a bed at right angles to the spindle axis, are available with spindle sizes from 100 mm. to 300 mm. dia. Tungsten carbide tooling can be employed to advantage on these machines, and a wide range of speeds and feeds is available.

This machine can be used for normal work on stationary workpieces, or with special fixed, sliding, revolving or swivelling tables, and special supporting standards.

OTHER FLOOR TYPES AVAILABLE

Portable, with short bed.

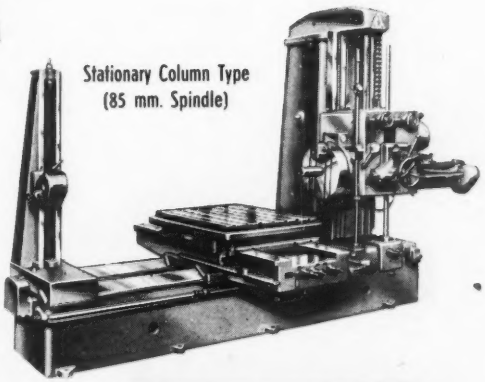
With revolving upright.

With swivelling headstock.

With revolving upright and swivelling headstock.

With revolving table travelling in the direction of spindle axis. Supporting standard adjustable in longitudinal and cross directions.

Stationary Column Type
(85 mm. Spindle)



STATIONARY COLUMN TYPE

Machines with the boring head mounted on a fixed upright are made in a range with spindle sizes from 69 mm. to 150 mm. dia. The table has longitudinal and cross travel, it can be rotated through 360° and indexed at 90° intervals. The supporting standard is adjustable in the direction of the spindle axis. Rigid construction ensures extreme accuracy and full advantage can be taken of tungsten carbide tooling. Speed and feed ranges cover an exceptionally wide field of applications, and single handwheel operation is applied to spindle speeds, all feeds, and all motions of the spindle and face-plate.

By employing the facing head (available as extra equipment) simultaneous boring and facing by face-plate and working spindle (each running separately) can be performed. The facing head slide can also be operated by hand during the boring operation with independent direction of feed.

WICKMAN *of* COVENTRY

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Wickman

172 F 34

The NEW Silentbloc FRUSTACON Mounting

This latest addition to the Silentbloc range has been designed to provide an engine mounting with a better performance at a lower price than any comparative type.

Of high load-carrying capacity and exceptional flexibility, the Frustacon has a range of deflection rates as wide as the engine designer could desire. The unusually large volume of rubber permitted by the Frustacon construction works in shear and compression. Compression pad (top) and rebound feature (bottom) are moulded in one piece.

The mounting is low in height, compact, easy to fit and readily accommodated in engine layouts.

with the right deflection for every load

Write for Technical Information

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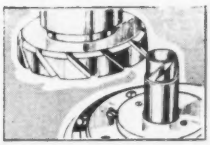
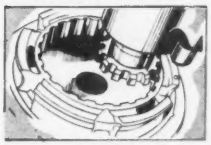
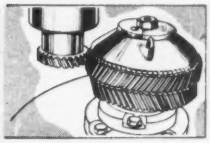
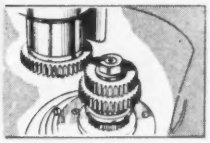
LORENZ

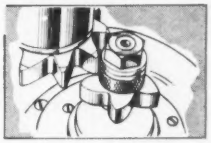
AUTOMATIC HIGH-SPEED GEAR SHAPING MACHINES

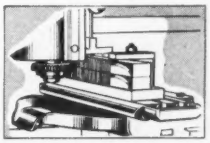


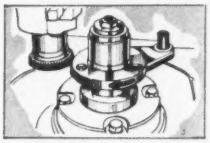


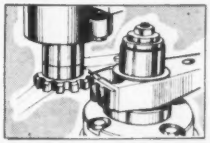


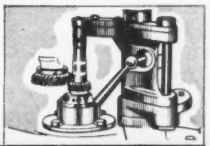


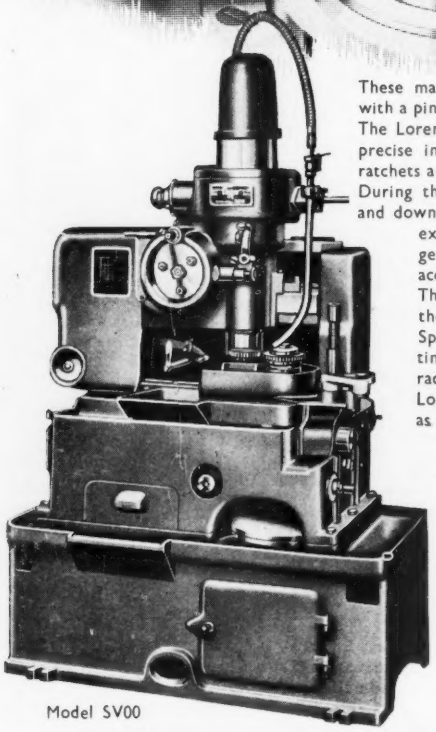












Model SV00

These machines shape by the generating process with a pinion type cutter for spur and helical gears. The Lorenz method is ideal for the production of precise internal and external gearing, sprockets, ratchets and other profiles.

During the cutting stroke the tool is moved up and down and rotated so that the tool and work execute the movement of two meshing gears. This method ensures maximum accuracy and smooth flanks of the work. The tool is re-sharpened by simply grinding the tooth face.

Special attachments are available for generating helical gears, gear trains, face gears, rack gears, shaping between centres, etc. Lorenz gear shapers are made in five sizes as shown below.

AUTOMATIC GEAR SHAPERS

MODEL	S 00	SV 00	S 5	S7 500	S7 1000
Pitch diam.	7⅛"	7⅛"	17⅞"	19⅞"	40"
Face width	2"	2"	3"	5⅛"	6.5/16"
Diam. pitch/ module	6/4	6/4	5/5	3,6/7	3,6/7

AUTOMATIC GEAR HOBBERS

High production machines of extreme accuracy for the manufacture of spur, helical, worm wheels, and special gearings are made for work of pitch diameters from 23⅛" to 63" and diam. pitch/module 4.23/6 to 1.59/16.

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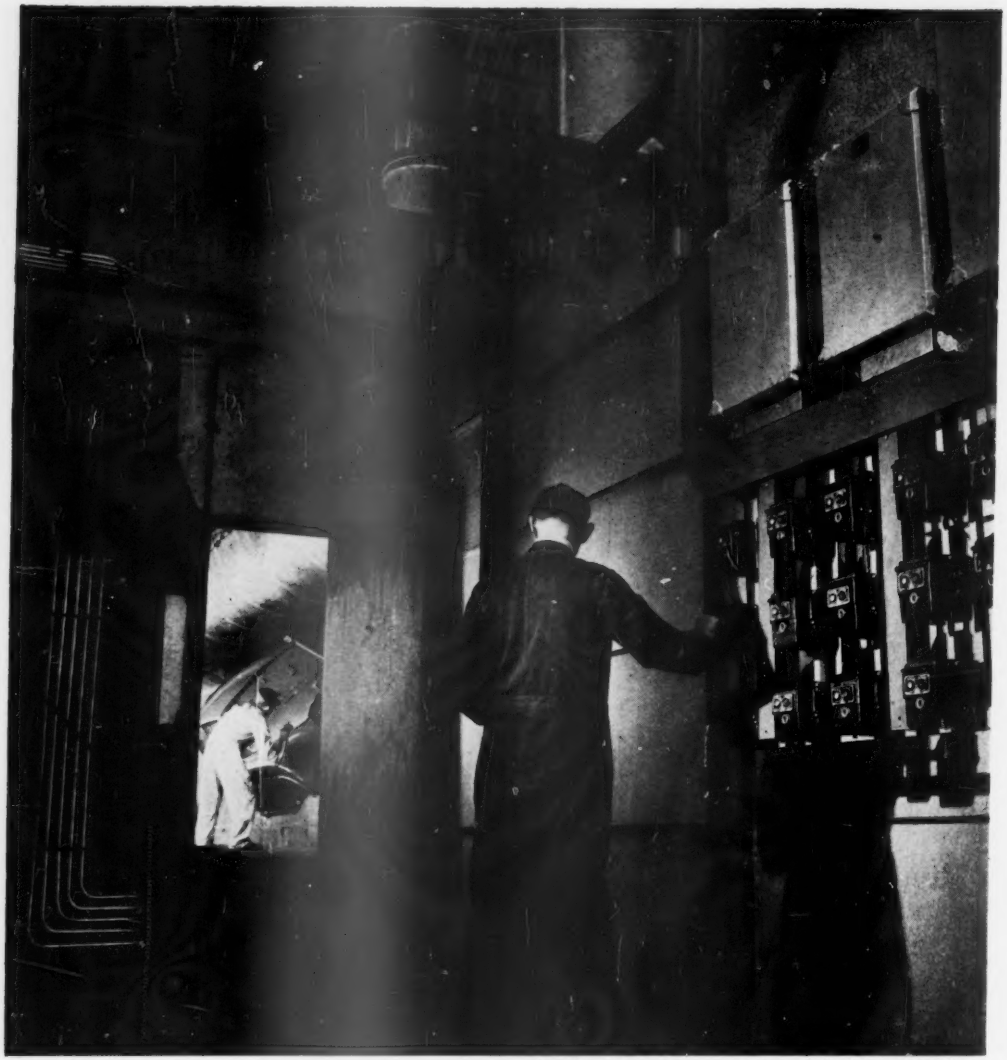

Air conditioning for car body spraying at Messrs. Fisher & Ludlow Ltd., Erdington

Does air conditioning affect output?

IN VERY MANY TRADES it does. Where consistent quality depends on constant temperature and humidity, air conditioning is the only way to ensure a low rejection rate. In others, where fumes or dust can mar the work, conditioned air is sometimes essential, and often desirable. Modern air conditioning plant provides predetermined air conditions in large or small spaces and is fully automatic. It is one more example of electricity used efficiently to maintain the level of production.

WHERE TO GET MORE INFORMATION

Your Electricity Board will be glad to help you to get the utmost value from the available power supply. They can advise you on ways to increase production by using Electricity to greater advantage — on methods which may save time and money, materials and coal, and help to reduce load shedding. Ask your Electricity Board for advice : it is at your disposal at any time.

Electricity *for* PRODUCTIVITY

Issued by the British Electrical Development Association

FASTER & BETTER TOOLS

REAMERS

Our specialist plant is at your service for the production of "REAMERS OF ALL TYPES" made from the F.B.T. "Speedicut" range of high speed steels, with "Speedicut SATIS" for the exacting demands of high tensile and work-hardening materials.

Extensive stocks of all standard types are available. Your special requirements for all purposes can readily be met.

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"MITIA" CARBIDE SAWS

SEGMENTAL SAWS
SAWS FOR ALL PURPOSES

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"HARDOMETER" HARDNESS TESTING MACHINE
TWIST DRILL POINT SHARPENING MACHINE
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AND ALL OTHER TYPES OF ENGINEERS' CUTTING TOOLS

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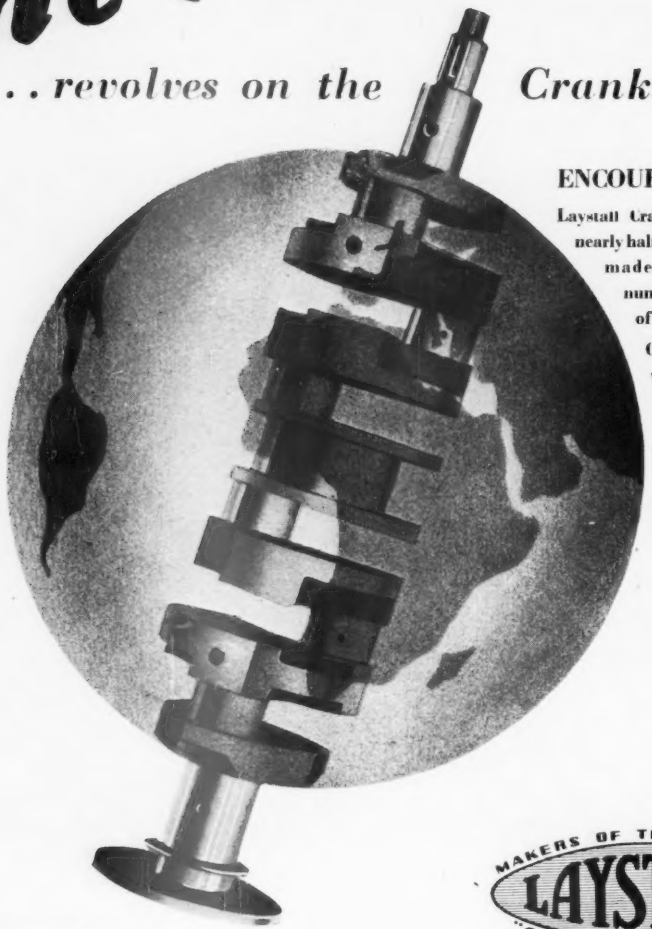
FIRTH BROWN TOOLS LTD.
SHEFFIELD

The I.C.* World

. . . revolves on the Crankshaft



ENCOURAGING NEWS!

Laystall Crankshafts—famous for nearly half a century—while being made in ever increasing numbers are still far short of the world demand.

Our new factory is working to capacity, but we hope, in the near future, to increase our output of the smaller marine and outboard types.

Enquiries are invited.

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★ *Internal Combustion.*

LAYSTALL ENGINEERING COMPANY LIMITED

Head Office: 53, Great Suffolk Street, London, S.E.1. **Telephone: WATerloo 6141**

Crankshaft Works: Green Lane, Aldersley, Wolverhampton. **Telephone: Wolverhampton 52006.**

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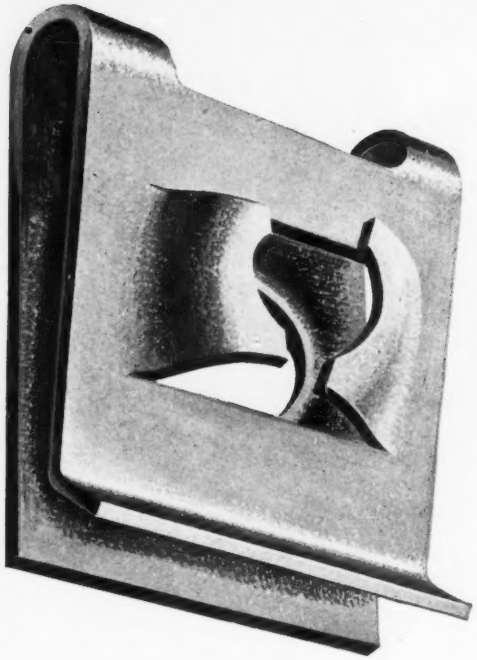

How are YOU fixed?

On the Triumph Mayflower the Standard Motor Co. Ltd. have saved labour and material costs by using 168 Spire Speed Nuts per car. Spire is the name. Speed is the reason. Spire Speed Nuts make faster, easier assembly — and that means saving. But not saving at the expense of performance. Far from it. These self-locking fasteners are functionally better than the old methods they replace. Each fastener saves 50/75% on weight of materials used.

114 SNU Spire Speed Nuts are used to secure Steering Column Attachment Bracket; Door Vent Drain Trough; Instrument Plate; Radiator Toggle & Clamp; Operating Lever; Handbrake Bracket; Dash Tray Millboards; Loudspeaker Attachment; Fascia Tray Cover; Steering Column Bracing Tube Bracket; Heater Blower Bracket; Trafficator Attachment; Bonnet Support Bracket; Driving Mirror; Sun Visors; Drain Channel Door Vent; Door Fittings; Rear Quarter Fittings; Rear Bumper Valance; Accelerator Pedal Bracket.

42 SNJ Spire Speed Nuts are used to secure Radiator Shell; Steering Column Sealing Plate; Steering Column Blanking Rubber; Heater Blower Attachment Plate; Heater Blanking Piece; Bumper Valances; Heater Duct; Dash Cover Plate.

5 SNP Spire Speed Nuts are used to secure Radio Loudspeaker Board; Speedo Trip (Slave).

7 SFP Spire Speed Nuts are used to secure Mayflower Badges; Door Armrests; Accelerator Pedal Bracket (to Dash).

Spire SPEED NUTS *fastest things in fastenings*

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Head Office & Works: Treforest, Glamorgan. Also Birmingham, Stockholm & Melbourne
CRC 255

CARRY ON BERRY

A Transport Slogan which held good in 1852
and still holds good in 1952

RICHARD BERRY & SON *Vehicle Suspension Specialists* **MAFEKING ROAD · SMETHWICK · STAFFS**

A
BROCKHOUSE
COMPANY

Private view

Here is a cut-away view of the Rockford clutch and power take-off, this clearly shows the balanced toggle-action which holds the clutch firmly in the engaged or the disengaged position without running thrust.

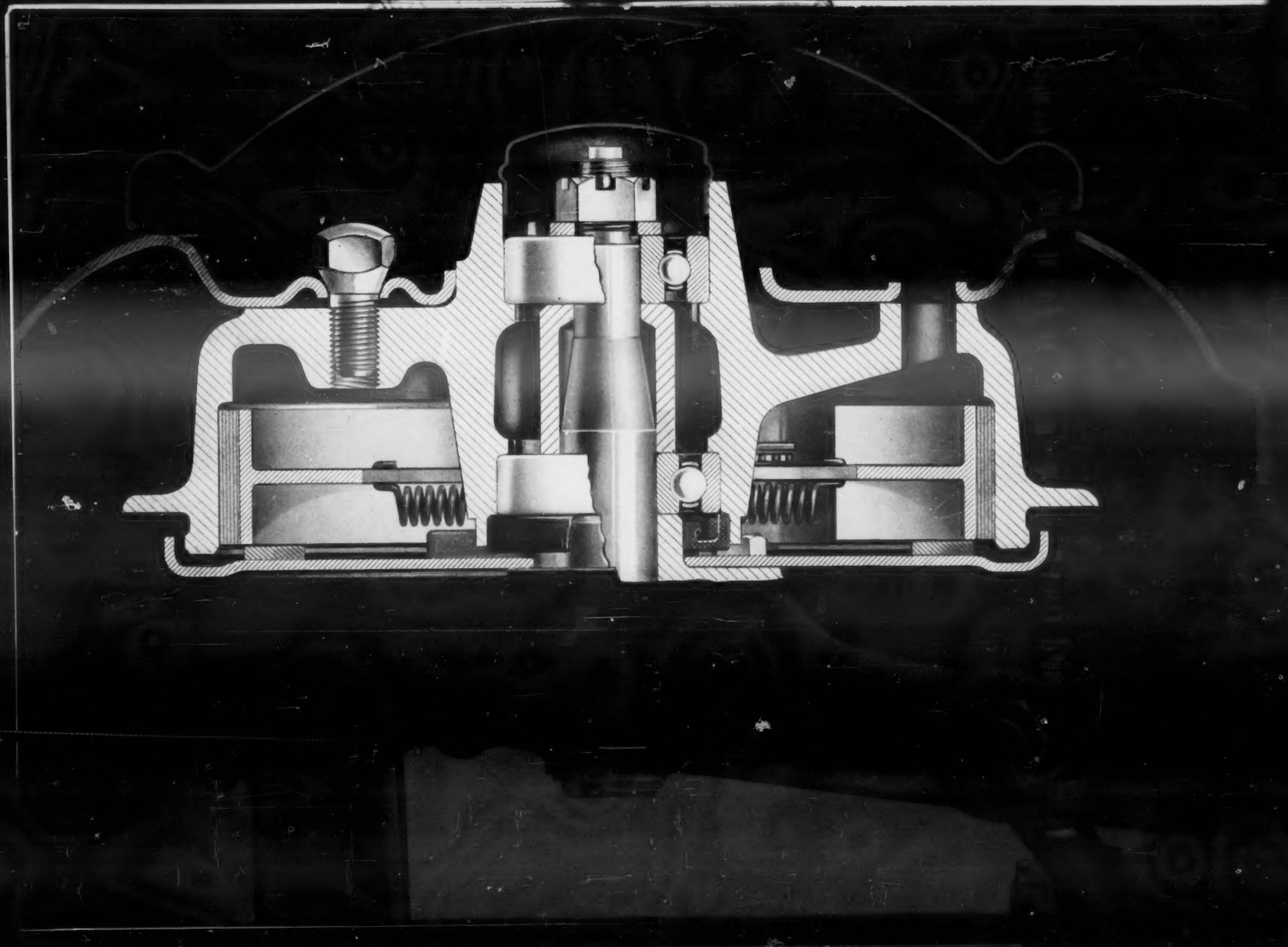
The extremely simple adjusting device is also visible: this is easily accessible and is self-locking without the need of special tools.

These units are in extensive use upon industrial petrol, paraffin and diesel engines, and upon portable compressors, power shovels, pumps, excavators, winches and cranes.

British made, in a range of sizes, by:—
BORG & BECK COMPANY LTD., LEAMINGTON SPA, ENGLAND

ROCKFORD

POWER TAKE-OFFS & CLUTCHES



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WE WILL COLLECT. Too small to be seen, too small to be stopped by ineffective filters, but nevertheless able to carry out vicious attacks on vital points.

It was to beat these enemies of durability that we developed the special plastic impregnated paper filters and as a result, they really do possess ....

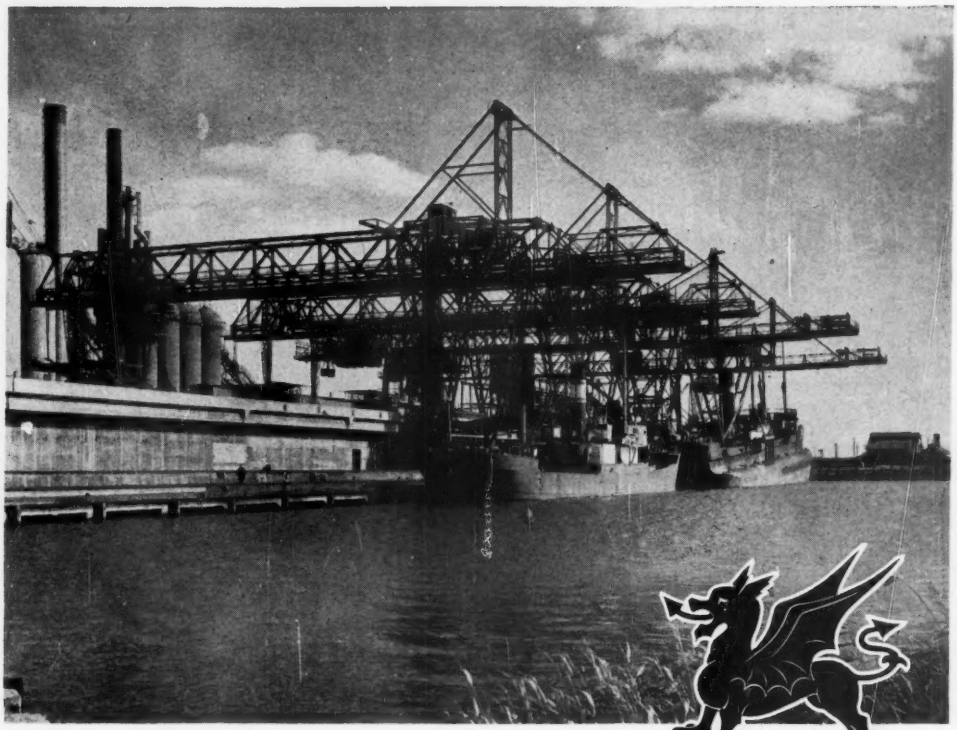
.... power of arrest

AUTOMOTIVE PRODUCTS COMPANY LIMITED, LEAMINGTON SPA.

PUROLATOR

REGD. TRADE MARK

MICRONIC *Oil* FILTER



Wharf at Margam Works showing Ore Unloaders

THE ORE UNLOADERS illustrated here can handle 500 tons of ore per hour. They are an important part of the new equipment which makes The Steel Company of Wales the most up-to-date plant of its kind.

PRODUCTS OF THE COMPANY ARE :—

Hot Rolled and Cold Reduced Sheets and Coils up to 72″ sheared width in qualities for auto-bodies, vitreous enamelling, extra deep drawing, etc. · Galvanized Flat and Corrugated Sheets · Electrical Sheets for Dynamos and Transformers · Blackplate · Hot and Cold Reduced Tinplate both hot dipped and electrolytically tinned · Plates ¼″ thick and upwards by up to 72″ wide · Flange, Bullhead and Bridge Rails · Fishplates · Steel Railway Sleepers · Sections · Slabs · Basic Pig Iron · Coke and By-Products · Sized Limestone.

* EXPORT SALES ORGANISATION FOR MILD STEEL FLAT ROLLED PRODUCTS . . . . .

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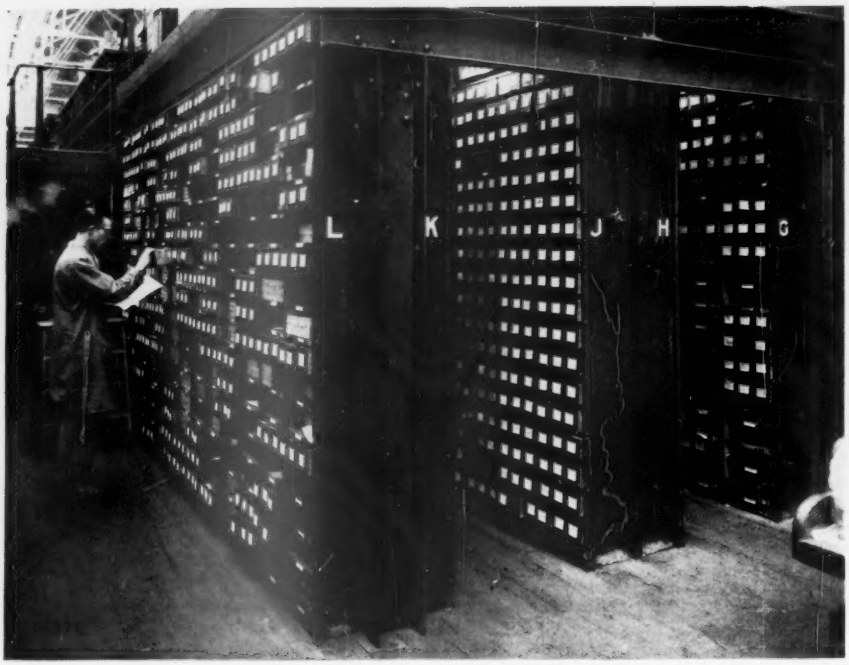
47 PARK STREET, LONDON, W.1.

THE STEEL COMPANY OF WALES LIMITED

ABBEY WORKS · PORT TALBOT

HERBERT



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WE have a large carefully selected stock of Coventry Dies, comprising all sizes and types in general use. Orders for these will be executed immediately.

Other dies can be delivered in 6 weeks from receipt of order.

Coventry double-tempered dies are made in ten types to suit the different materials now in use, including the alloy steels that are difficult to cut.

We guarantee to supply dies which will cut clean, accurate threads on any material that can be threaded.

PLEASE SEND ENQUIRIES TO THREADING DEPT.

TELEPHONE 88781. EXTENSION 210.

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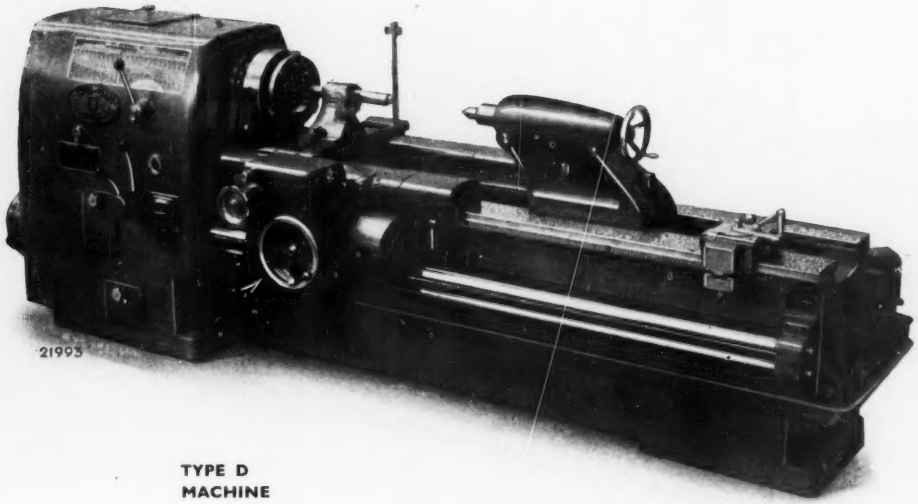

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MACHINE

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HIGH-SPEED THREADING MACHINES

These machines, which are semi-automatic, will cut external, internal, right- or left-hand, parallel, taper, single- or multi-start threads of various forms and pitches in all materials, quickly, cheaply and accurately.

Setting up is simple; design facilitates the full use of single-point Ardoloy or other tungsten-carbide tipped tools.

Precision threads can be cut in a fraction of the time taken on thread milling machines or centre lathes.

Two sizes for cutting threads up to 4in. and 12in. diameter.

	Type B.	*Type D.
Maximum threading diameter, external	4″	12″
" " " internal	6″ approx. (short pieces only)	16″
Normal threading length	$2\frac{5}{8}$″	$5\frac{1}{2}$″
Length between centres	3′	5′
Maximum number of starts	12	12
Maximum swing over bed	13″	21″
Motor	5 h.p.	14 h.p.

***EARLY DELIVERY**

Full details from Factored Division, Red Lane Works. *Phone:* 89221 (10 lines)

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Grooved Pins
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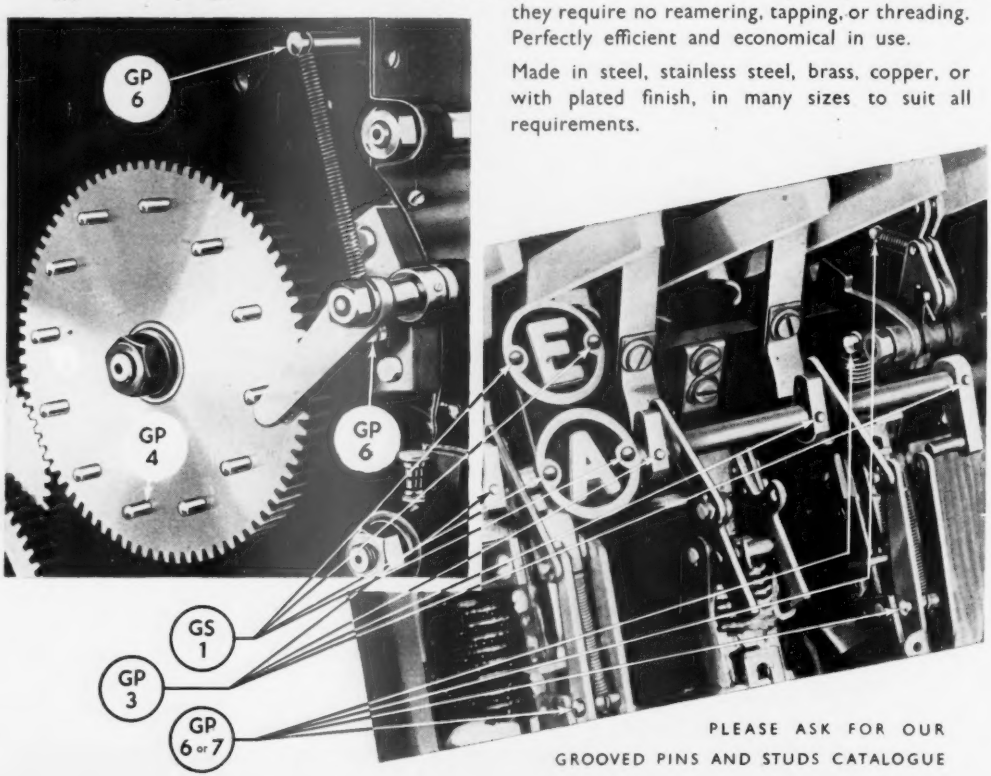
Drill a hole with a standard drill and drive in a Grooved Pin — no reamering

Mills Grooved Pins and Studs have proved themselves in many branches of the engineering and allied trades. Vibration proof when driven into a plain parallel hole, they require no reamering, tapping, or threading. Perfectly efficient and economical in use.

Made in steel, stainless steel, brass, copper, or with plated finish, in many sizes to suit all requirements.



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For longer life *and* greater efficiency

There are "material" reasons why HAGCO "Silmocrom" and "Austenitic" Cylinder Liners are "miles better". During the casting on the special HAGCO Spun-casting machine, the whole assembly spins at high speed. Centrifugal force holds the molten iron to the mould, whilst the impurities, being lighter in weight than the iron, are forced to float on the inner surface of the metal and are flushed away before the casting solidifies. The result is that HAGCO Cylinder Liners are true *right through* —and ensure longer and more efficient cylinder life.

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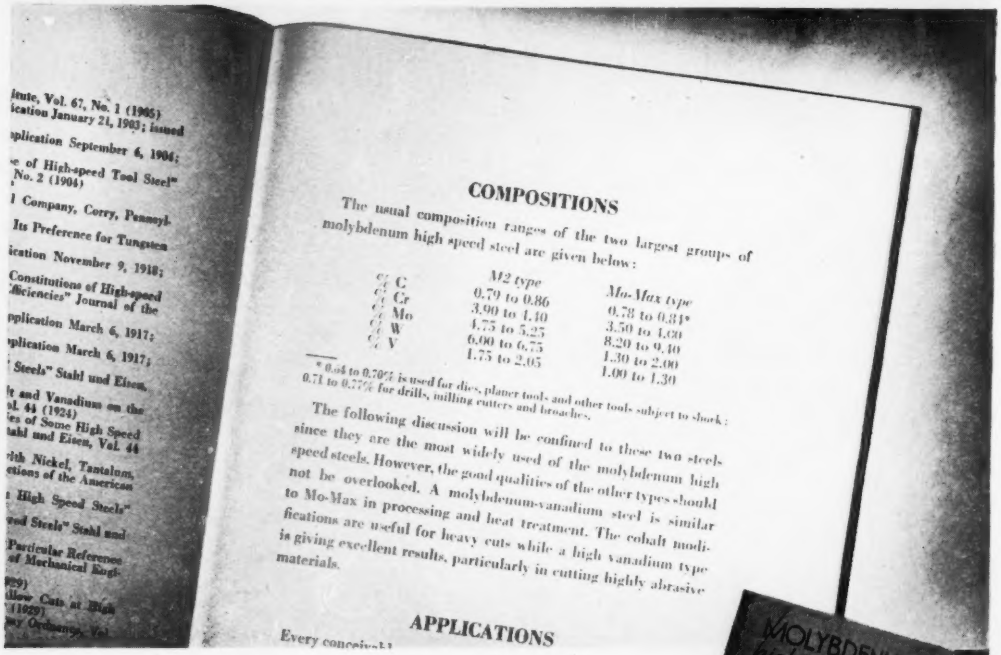
MOLYBDENUM HIGH SPEED STEELS



...itute, Vol. 67, No. 1 (1905)
...ication January 21, 1903; issued
...plication September 6, 1904;
...e of High-speed Tool Steel"
...No. 2 (1904)
...l Company, Corry, Pennsyl-
...Its Preference for Tungsten
...ication November 9, 1918;
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...plication March 6, 1917;
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...dium Cuts at High
...(1929)
...per Ordnance, Vol.

COMPOSITIONS

The usual composition ranges of the two largest groups of molybdenum high speed steel are given below:

	M2 type	*Mo-Max type*
% C	0.79 to 0.86	0.78 to 0.31*
% Cr	3.90 to 4.10	3.50 to 4.00
% Mo	4.75 to 5.25	8.20 to 9.40
% W	6.00 to 6.75	1.30 to 2.00
% V	1.75 to 2.05	1.00 to 1.30

* 0.54 to 0.70% is used for dies, planer tools and other tools subject to shock; 0.71 to 0.77% for drills, milling cutters and broaches.

The following discussion will be confined to these two steels since they are the most widely used of the molybdenum high speed steels. However, the good qualities of the other types should not be overlooked. A molybdenum-vanadium steel is similar to Mo-Max in processing and heat treatment. The cobalt modifications are useful for heavy cuts while a high vanadium type is giving excellent results, particularly in cutting highly abrasive materials.

APPLICATIONS

Every conceiv...

— WILL WIN ON MERIT

Despite the present difficult situation, make no mistake, Molybdenum High Speed Steels with their superior performance will eventually come out on top.

Some of the reasons why are given in this 30 page Technical Booklet, which deals with compositions, applications, performance and treatment.

Write now for a copy. It is available free on request to tool manufacturers and users.

Climax Molybdenum Company of Europe Ltd.

MOLY

Technical enquiries for this book, preferably with full details of the projected application of Molybdenum steel, should be addressed to:

99 Pinstone St. Sheffield, 1.

(*Registered Office :* **2-3 Crosby Square, London E.C.3.**)

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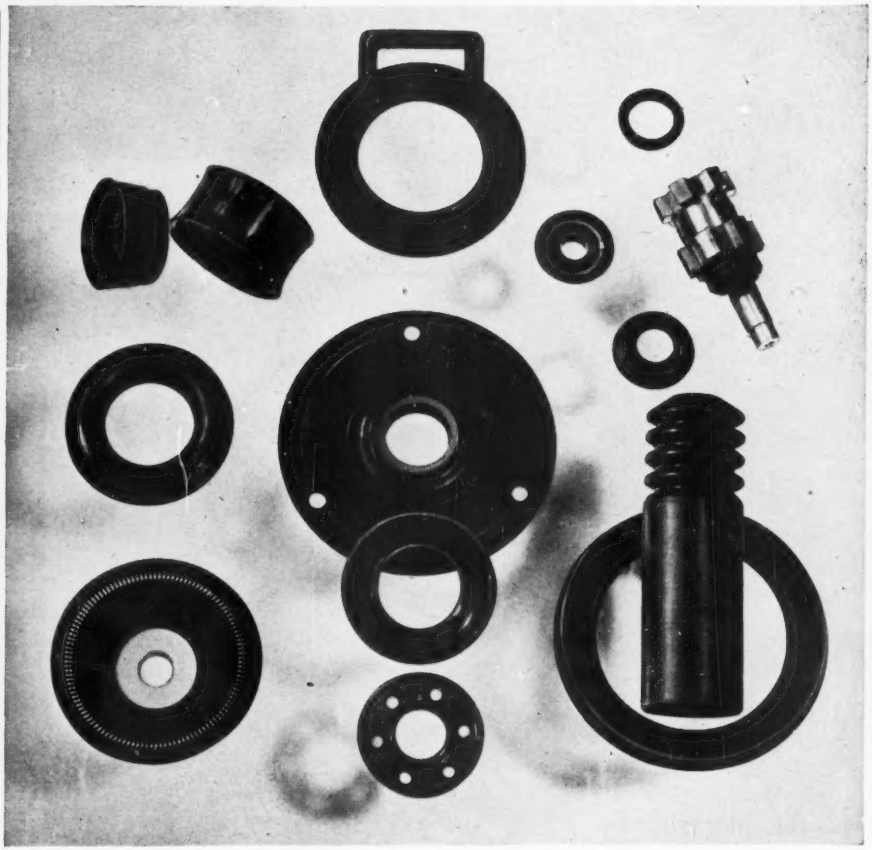

"Gaco" is *ideal* for hydraulic sealing. It is grainless, resilient, non-porous, highly resistant to Oils, Fuels and Solvents, and has excellent wearing properties. "Gaco" mouldings are produced in a great variety of shapes and hardnesses and are well-known for their superior surface finish and dimensional accuracy.

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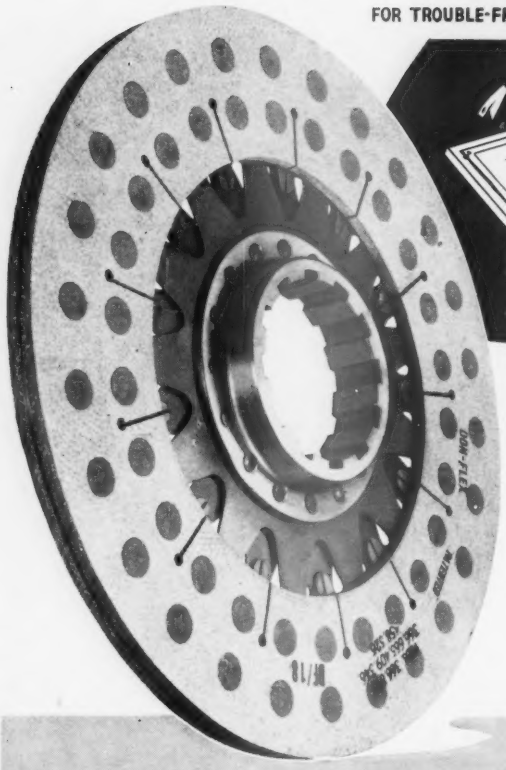

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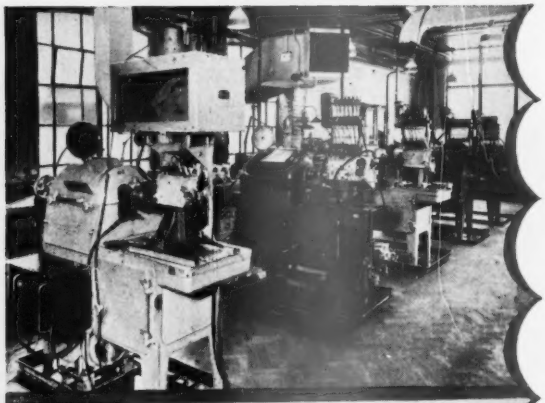

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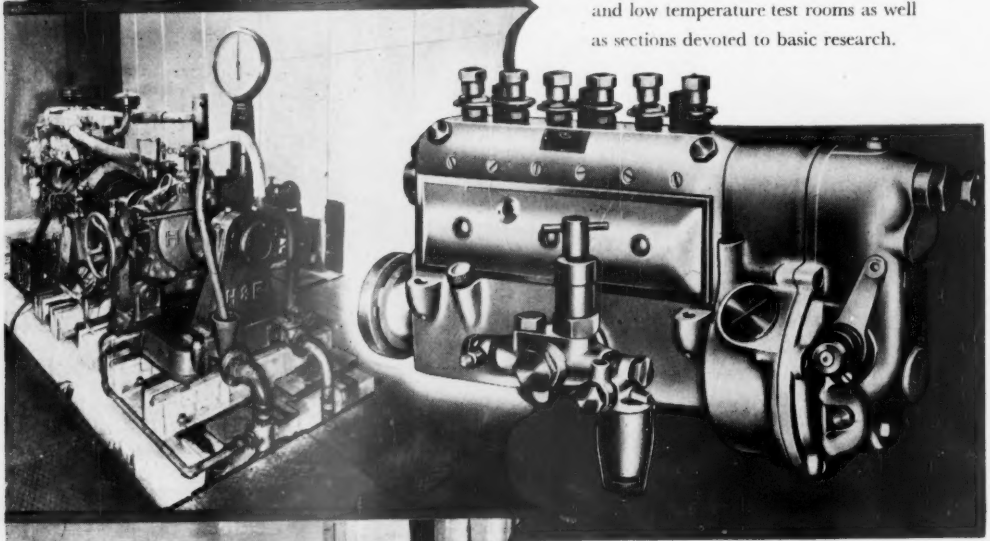

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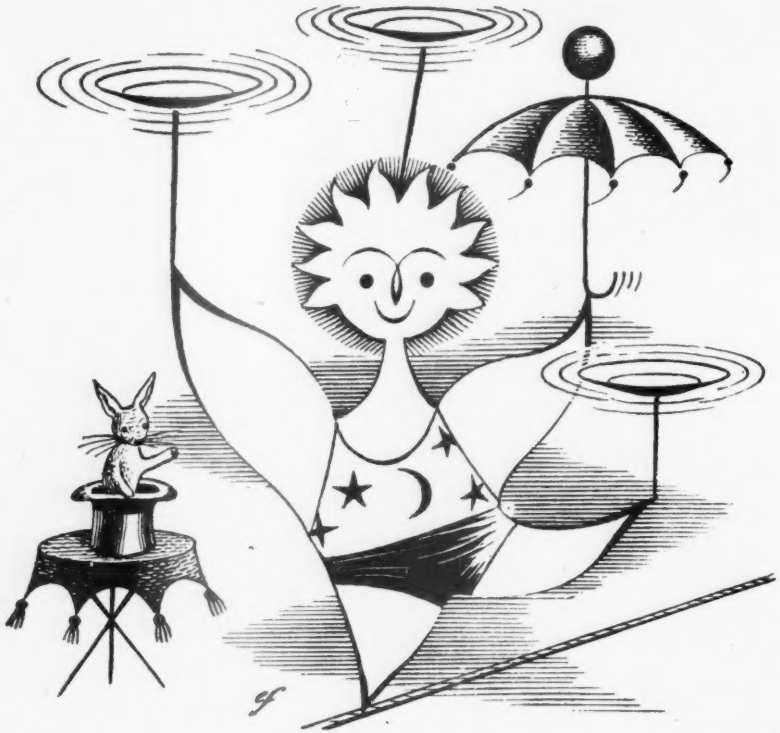

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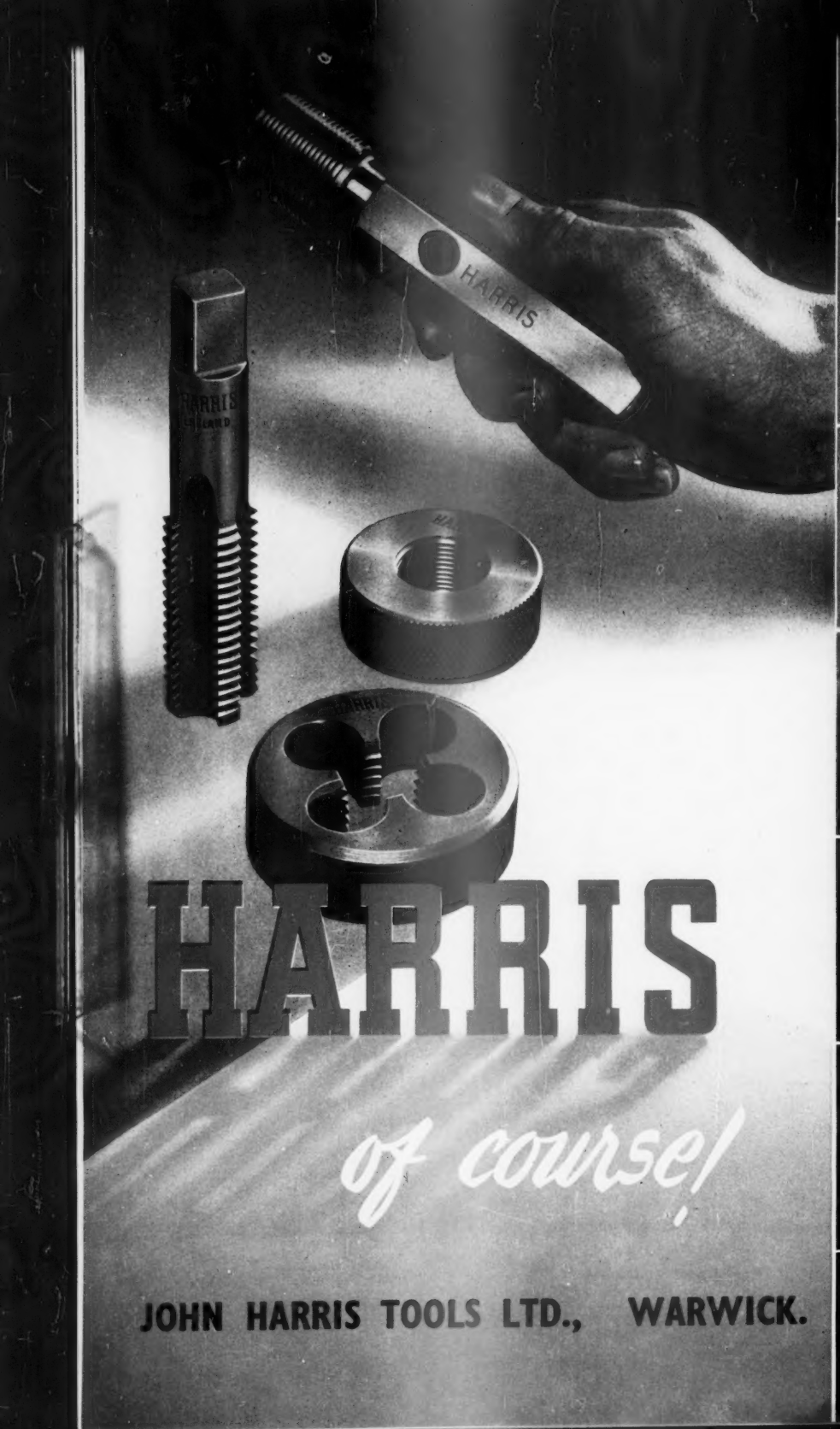

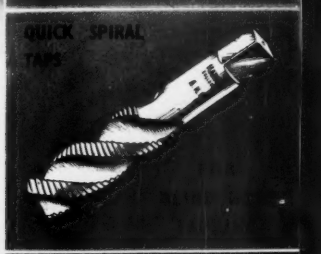

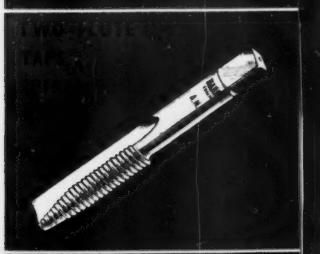

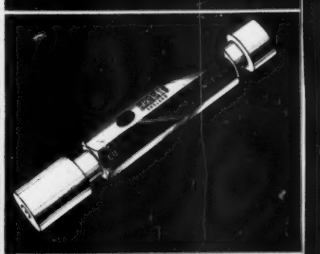

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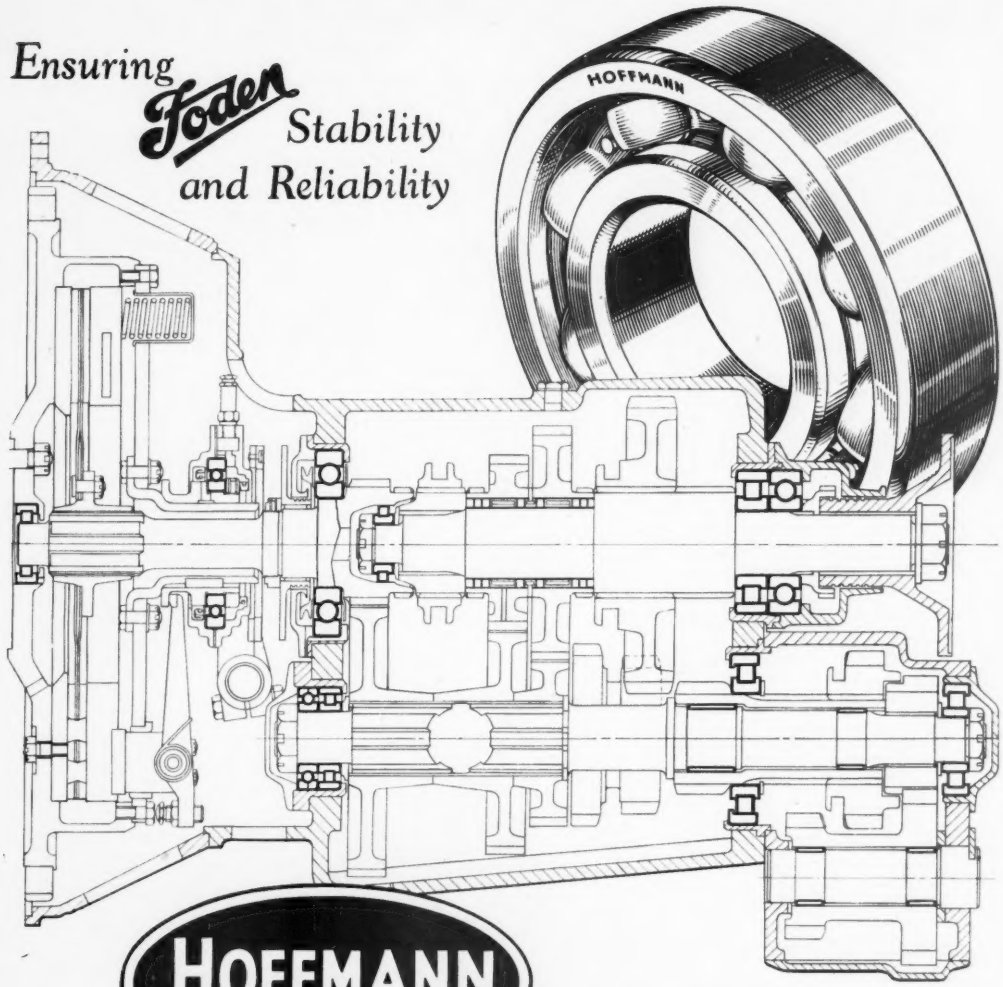

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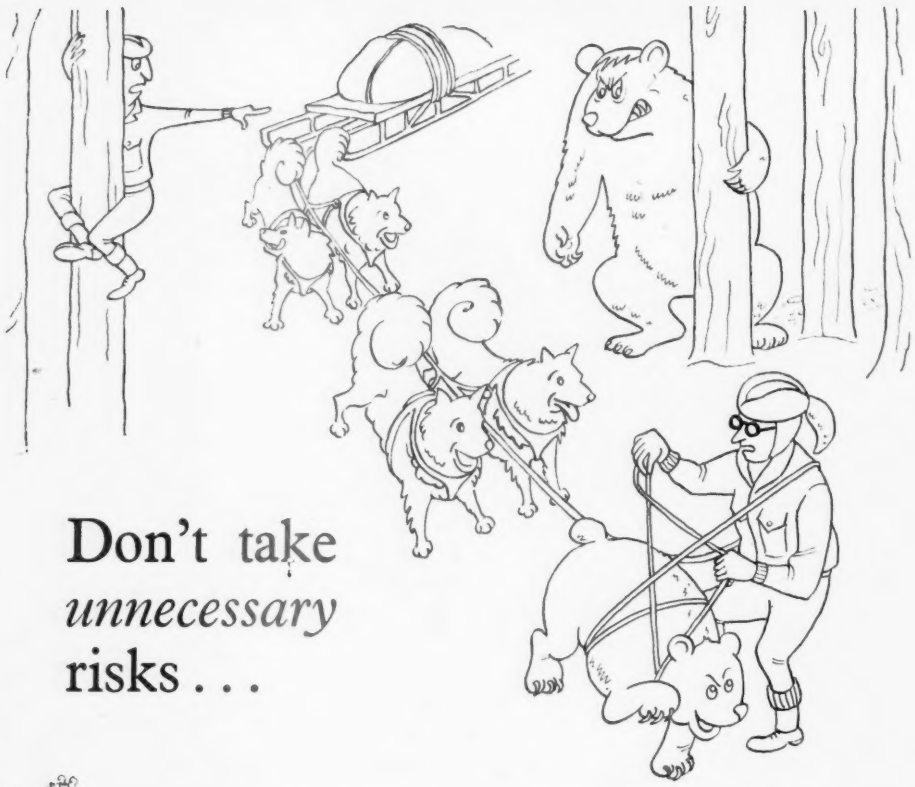

Don't take *unnecessary* risks . . .

ONLY a boor would upset a bear, but at some time or other the smartest of trappers is apt to get trapped. From farthest North to farthest South, in ice-bound pools and fog-bound factories, someone or other somewhere or other is going to take that one false step — that one unnecessary risk — that spells Trouble with a capital T. And if ever he finds his way out of the wood, he'll take cold comfort from the fact that he's only got himself to blame, and he'll be wondering: Was it worth it . . . ?"

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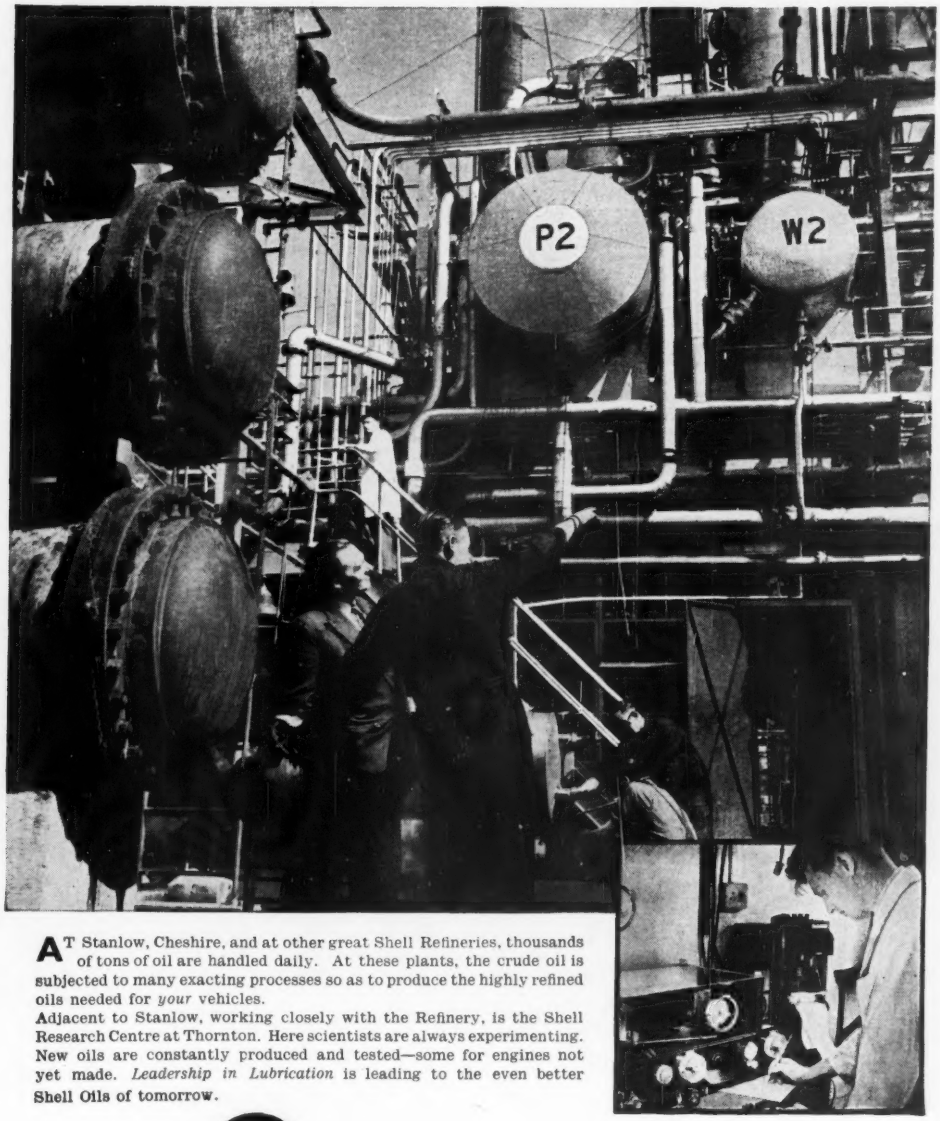

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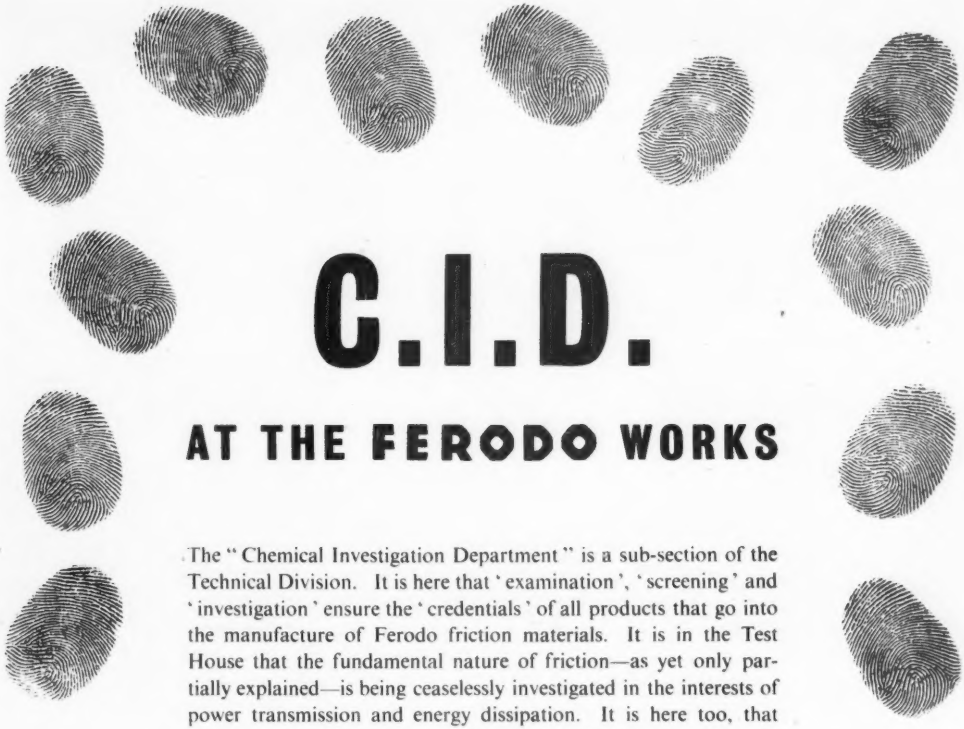

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AT THE FERODO WORKS

The "Chemical Investigation Department" is a sub-section of the Technical Division. It is here that 'examination', 'screening' and 'investigation' ensure the 'credentials' of all products that go into the manufacture of Ferodo friction materials. It is in the Test House that the fundamental nature of friction—as yet only partially explained—is being ceaselessly investigated in the interests of power transmission and energy dissipation. It is here too, that samples of every batch of output are subjected to drastic endurance and efficiency tests. But it is on the M.I.R.A. track (of which Ferodo Limited are one of the largest users) that the final and practical proving of the finished product takes place.

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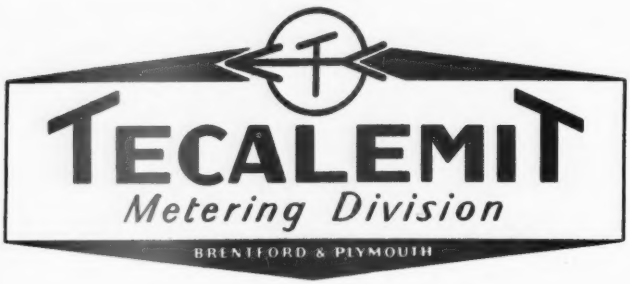

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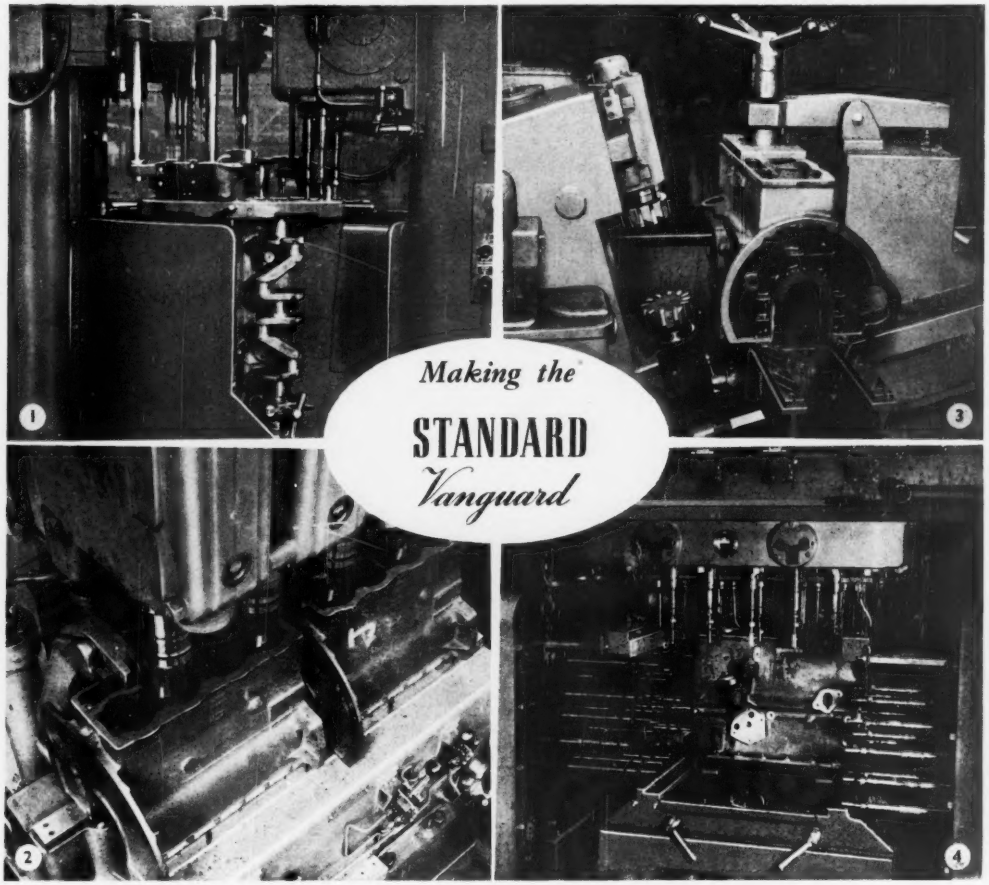

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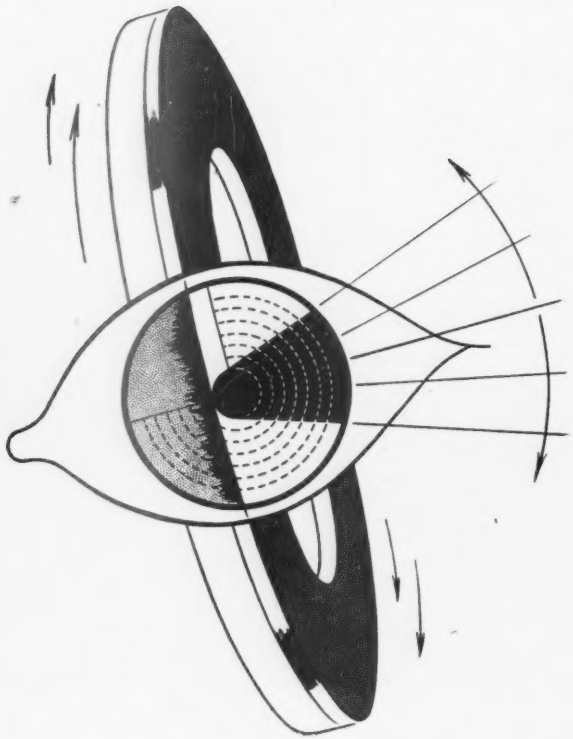

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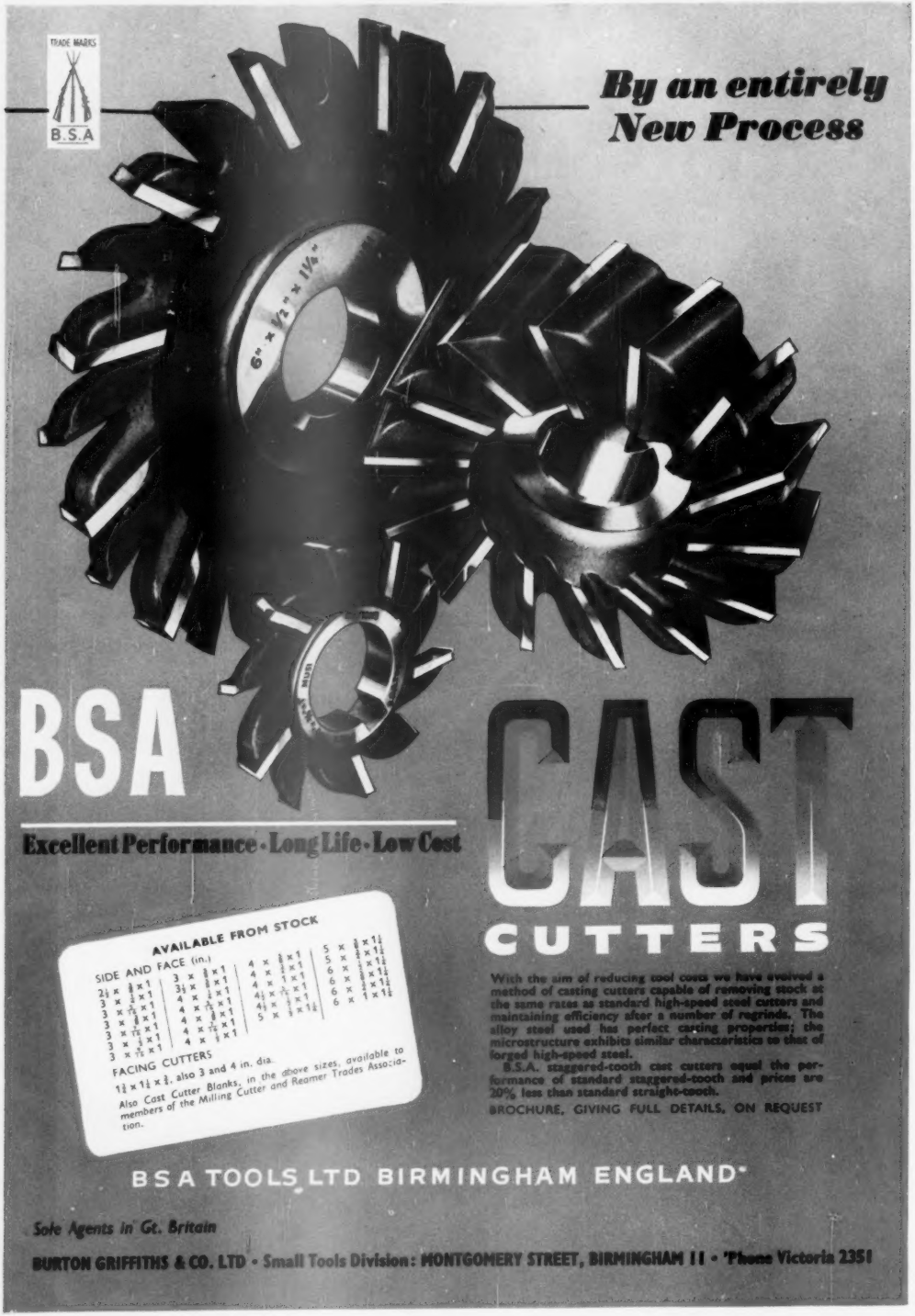
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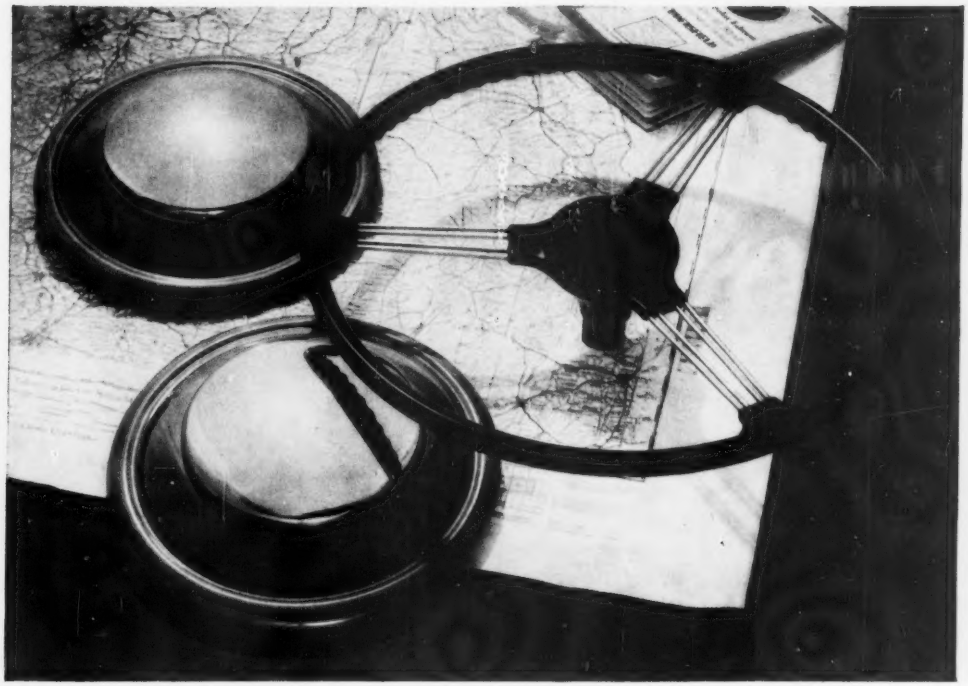

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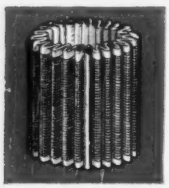
That is the vital question which must be asked when considering filtration claims. Engineers today accept the fact that for real protection, partial flow filtration is not enough. But remember! even a so-called *full* flow filter can allow irreparable damage if the by-pass comes into operation for *long* periods when the oil is cold.



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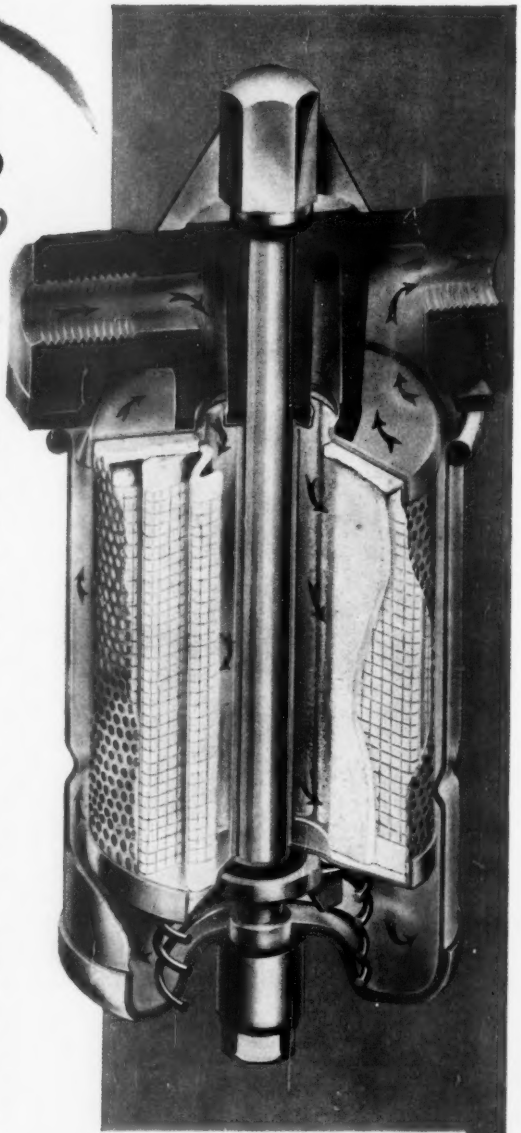
Here is the VOKES filter element — unchanged in principle because thirty years of constant research have found nothing better. By its principle of "filtration-in-depth" it avoids the possibility of particles being driven straight through the filtering medium and also ensures a slower build up of surface sludge.

This then is the vital point. If a so-called full flow filter cannot cope with the oil when it is cold and sends it direct into the engine, then it is not filtering *all* the oil *all* the time. Consider, too, the question of "cleanability". This is highly important where plant and machinery is operating in remote areas, avoiding the possibility of equipment being put out of service through the non-availability of expendable elements. Also, although the VOKES element may cost a little more in the first place, the fact that it can be cleaned several times gives long-term economy.

VOKES

Pioneers of scientific filtration

V.61.



The VOKES direct flow device is a safeguard to the engine and not just to the filter itself. It comes into operation only under *abnormal* conditions and *not* for long periods merely because the oil is cold. When it is called upon, however, it is truly foolproof. Illustration (left) shows flow through filter element. Illustration (right) shows by-pass flow.

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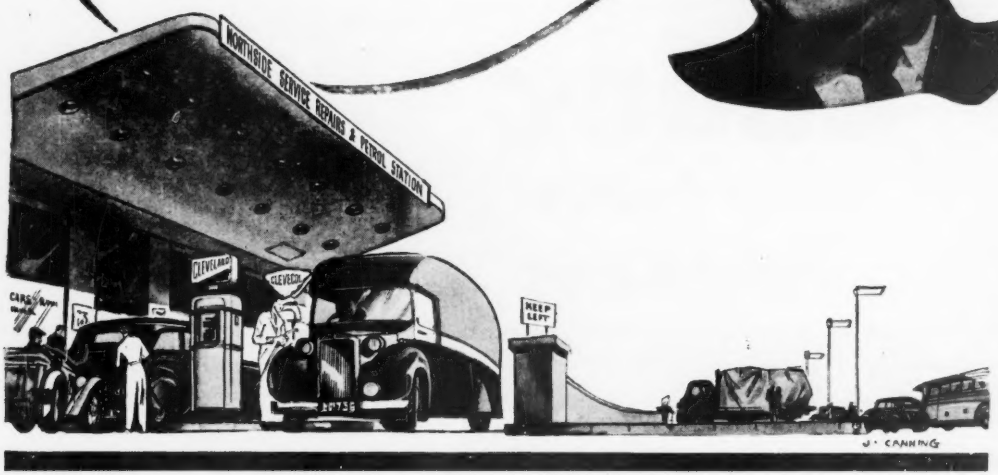

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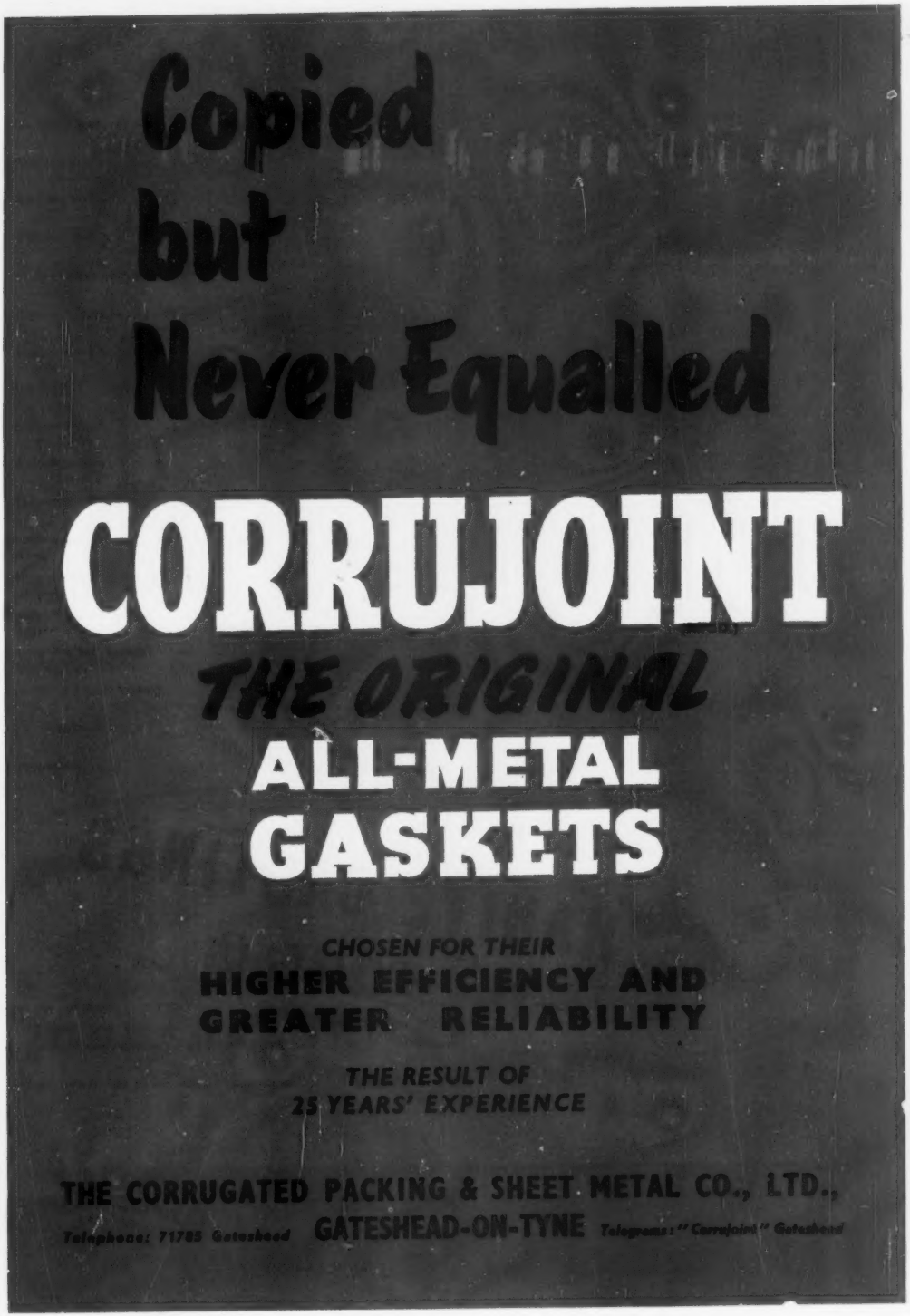
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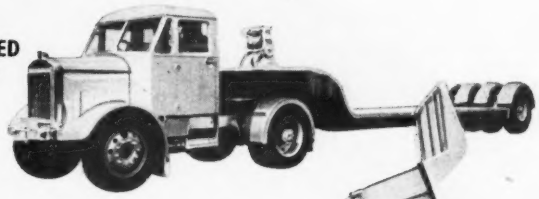

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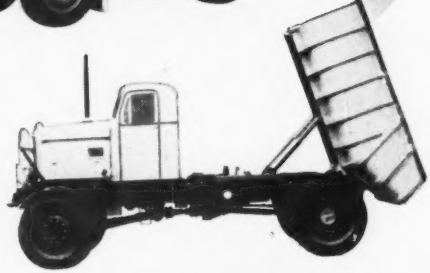

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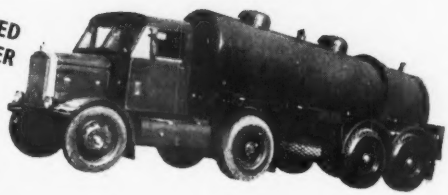

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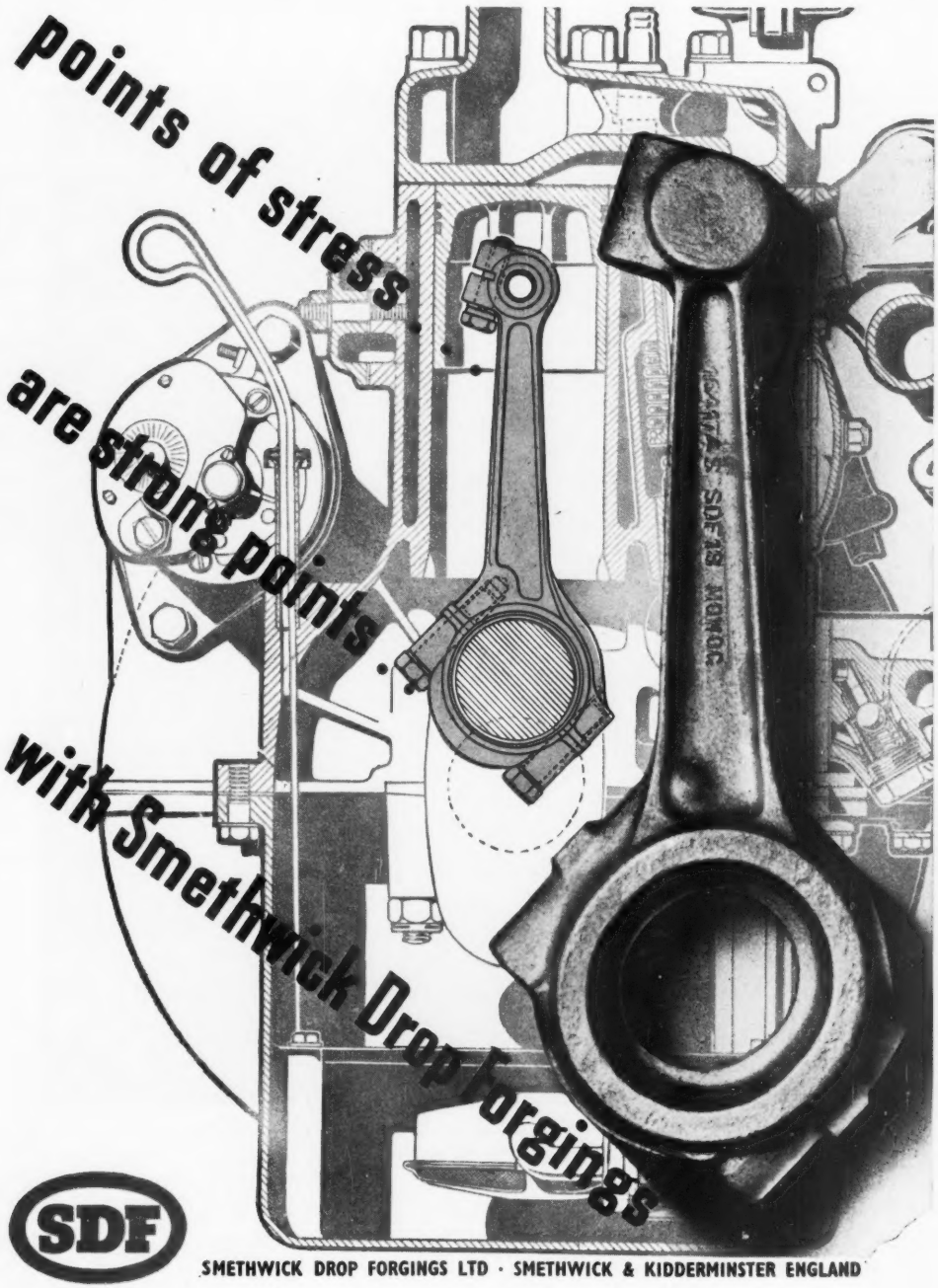
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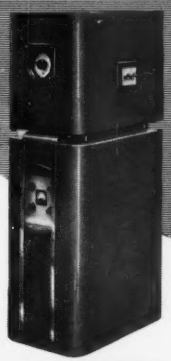

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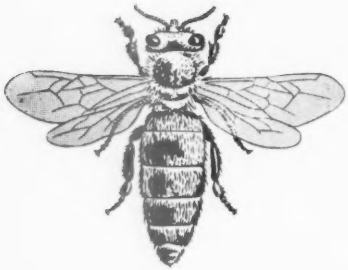

There are something like 24,000 species of bees. Fat bees, thin bees, hairy bees, bald bees, black bees, brown bees, and bees with spots. Their range of development varies considerably and it has been found that the less developed species of bees have a short labinum which only allows them to take nectar from shallow plants. Bees — like the honey bee and bumble bee — decent types, have long labina, allowing them to reach into the deepest flowers. It's all a question of breeding — those with the highest breeding are perfectly designed to do the job.

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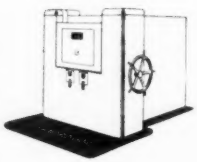

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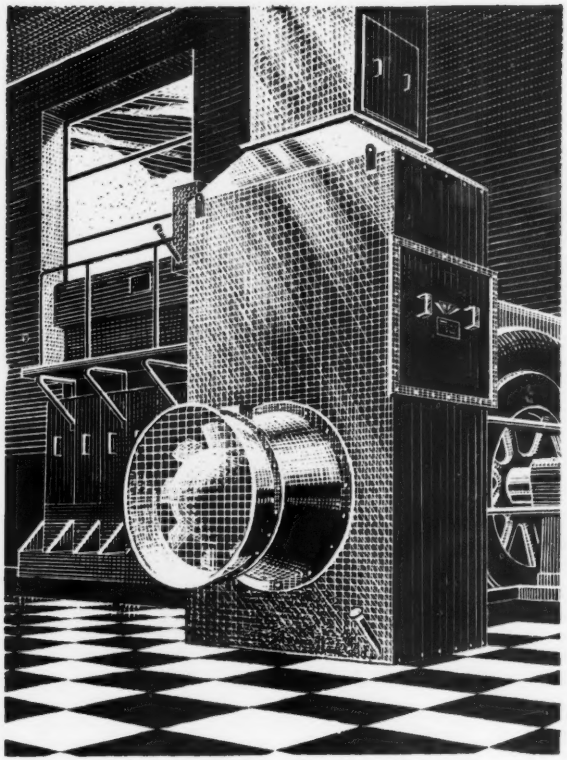

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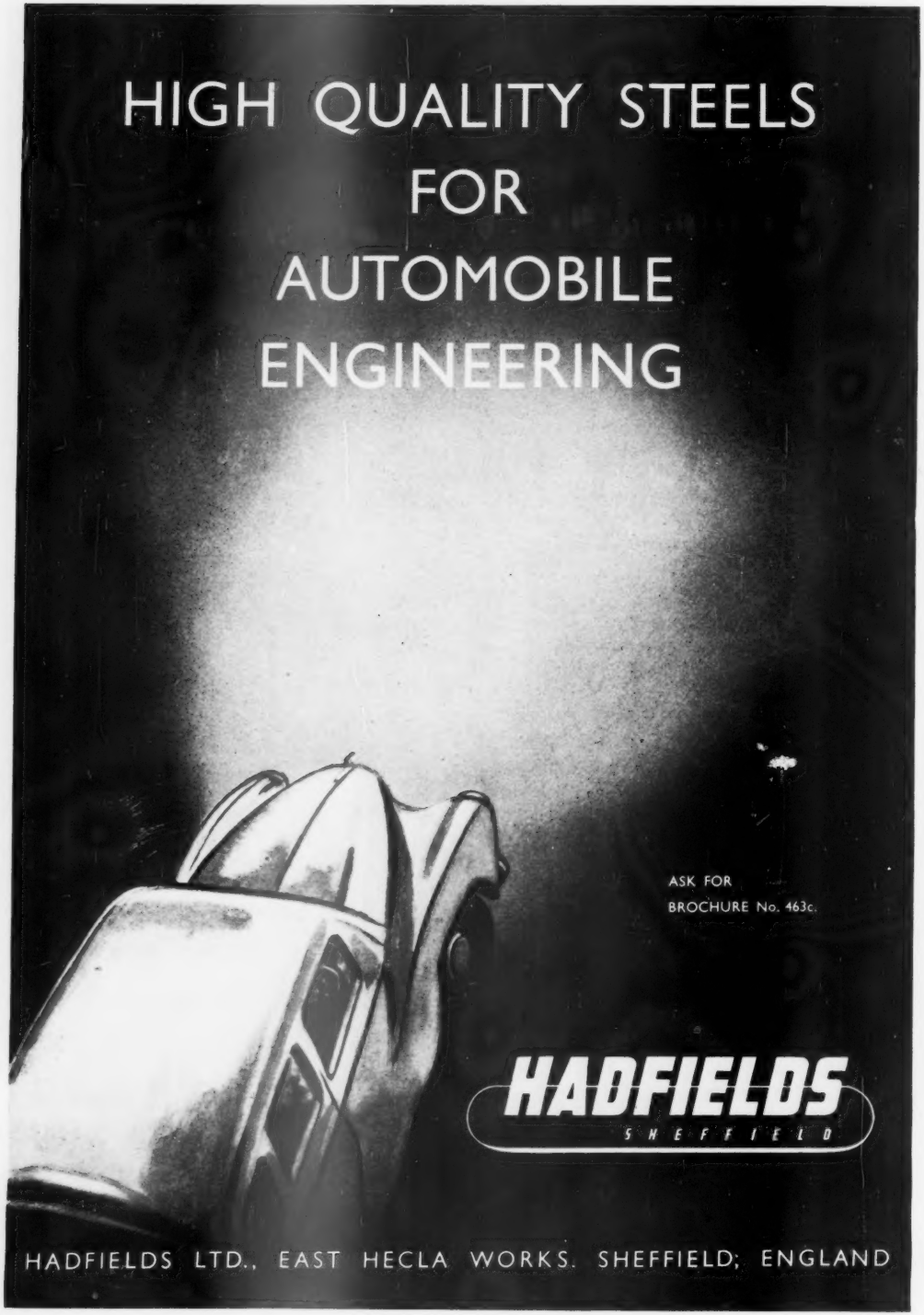
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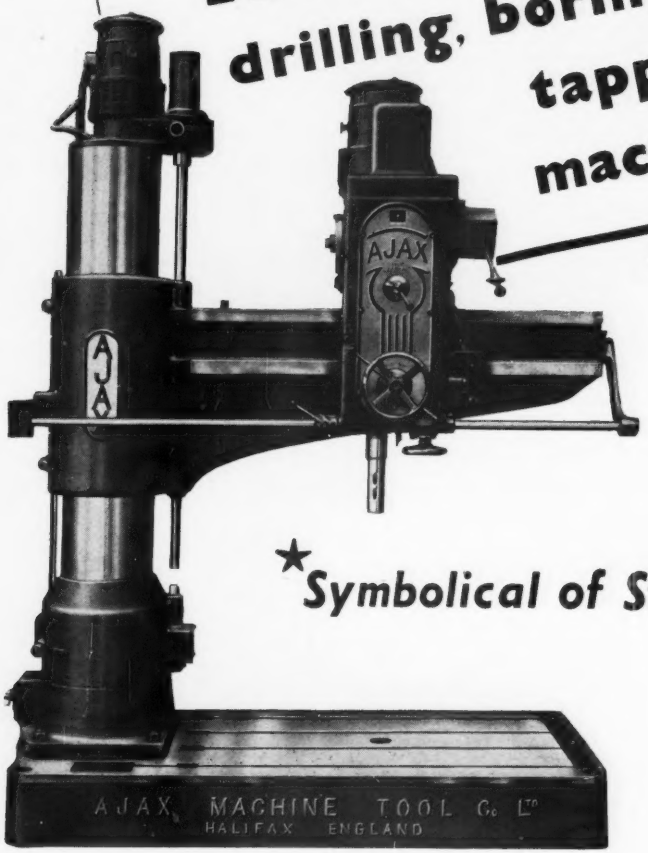

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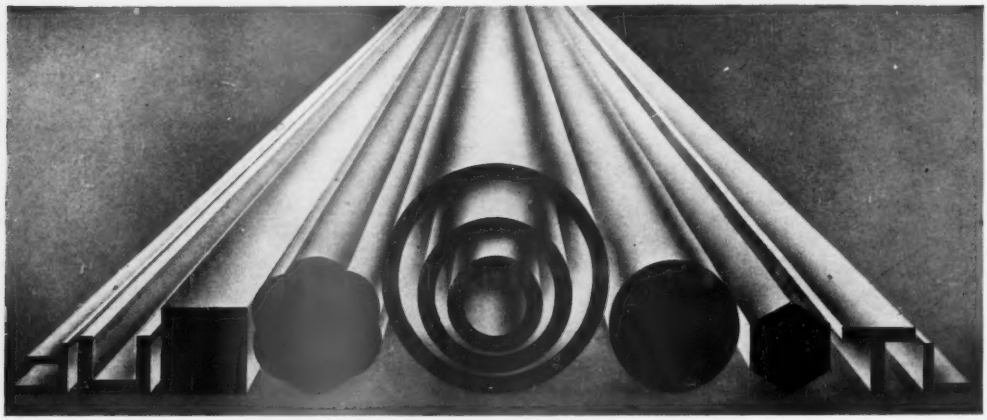

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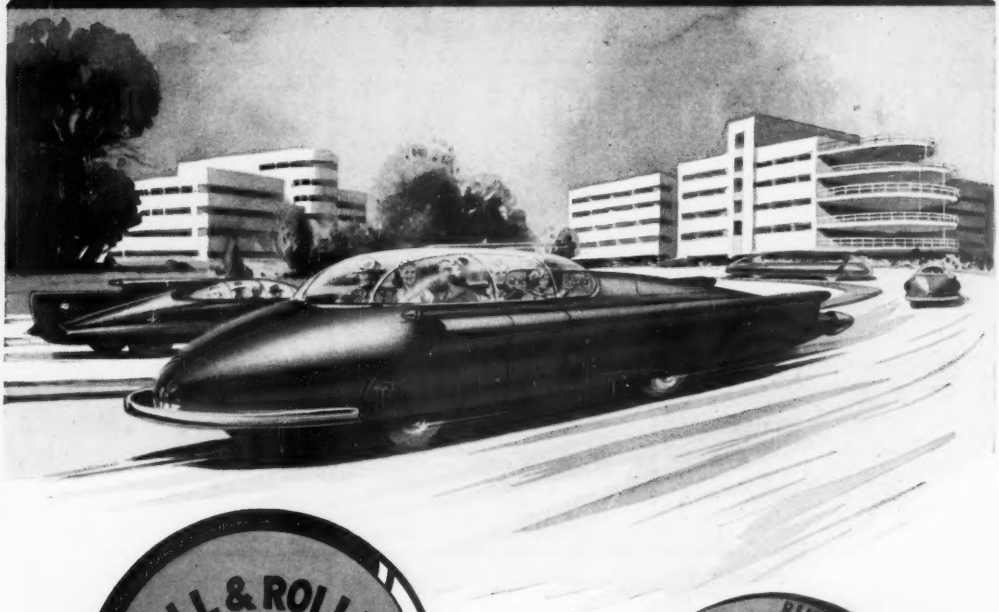

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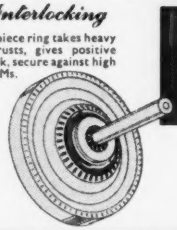
2-piece ring takes heavy thrusts, gives positive lock, secure against high RPMs.



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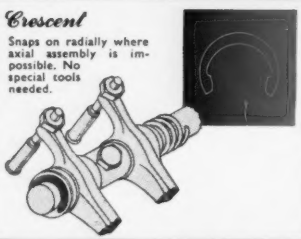
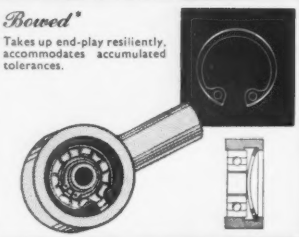
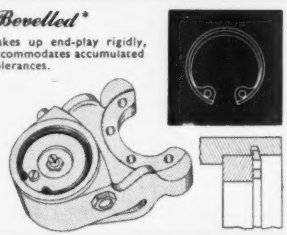


*Bowed**
Takes up end-play resiliently, accommodates accumulated tolerances.



*Bevelled**
Takes up end-play rigidly, accommodates accumulated tolerances.



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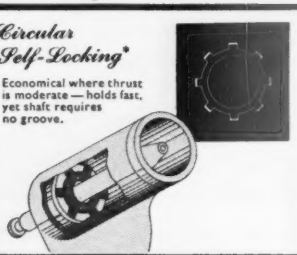
Economical where thrust is moderate — holds fast, yet shaft requires no groove.



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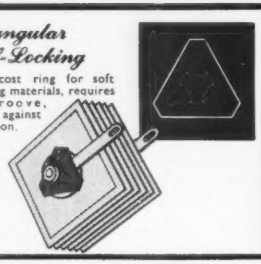
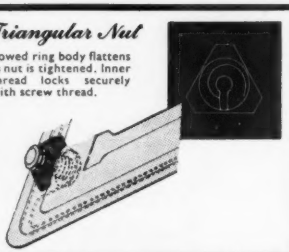
Low cost ring for soft shafting materials, requires no groove, secure against vibration.



Triangular Nut
Bowed ring body flattens as nut is tightened. Inner thread locks securely with screw thread.



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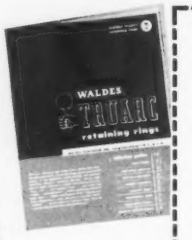

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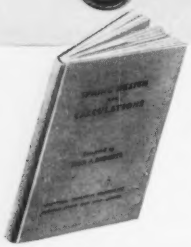

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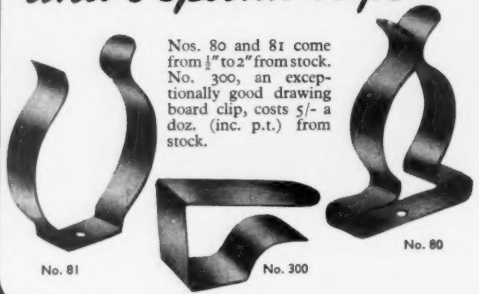

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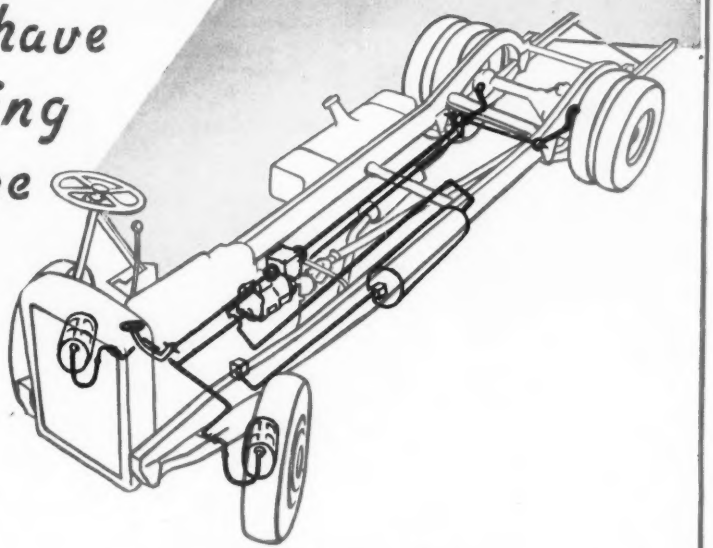

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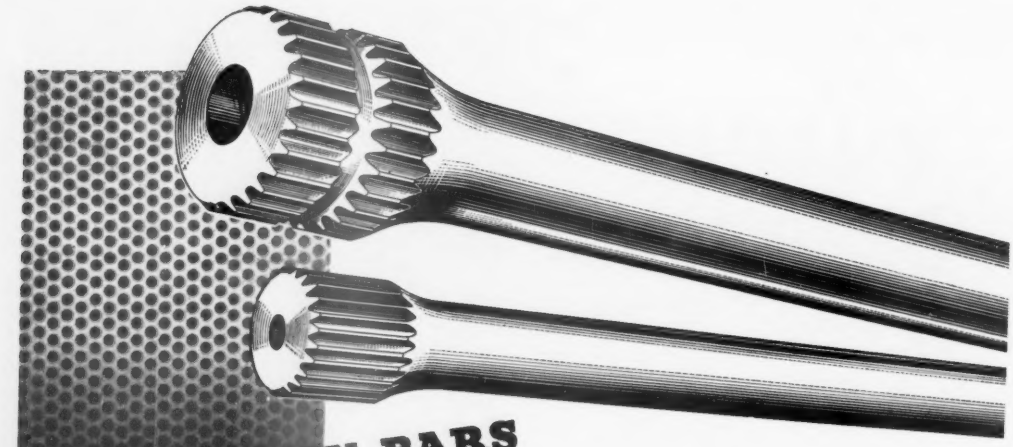

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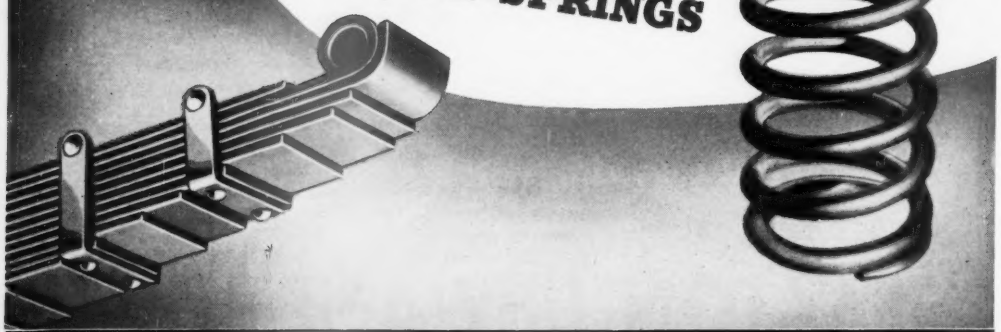

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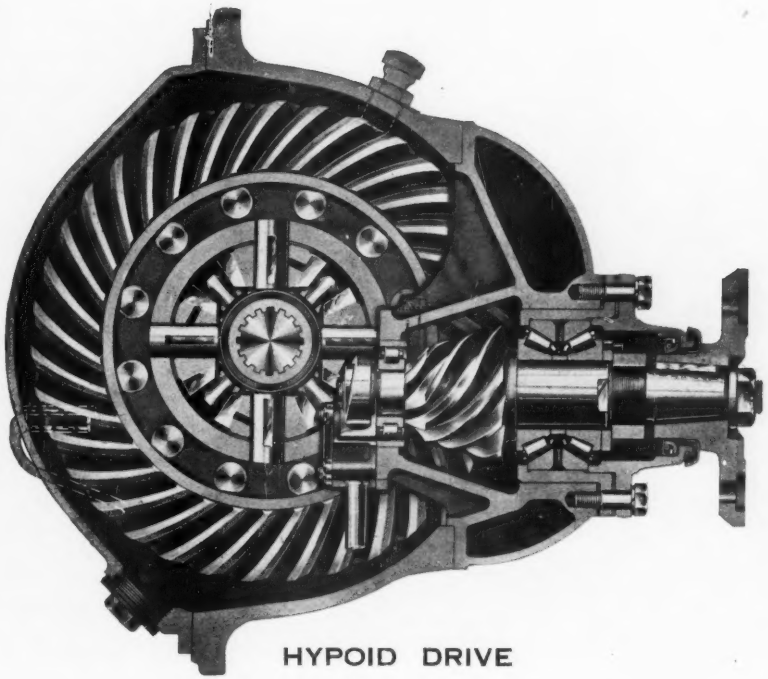

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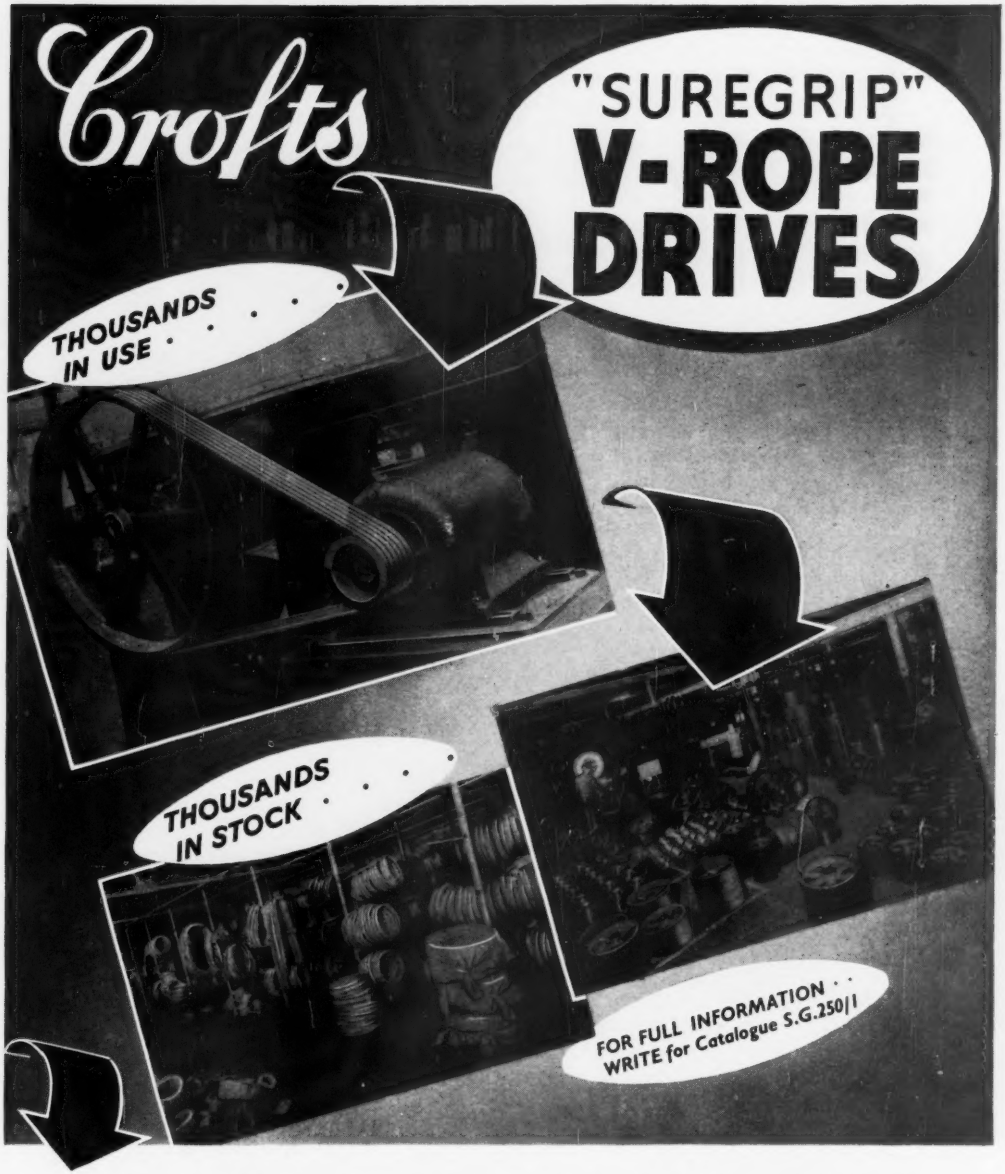

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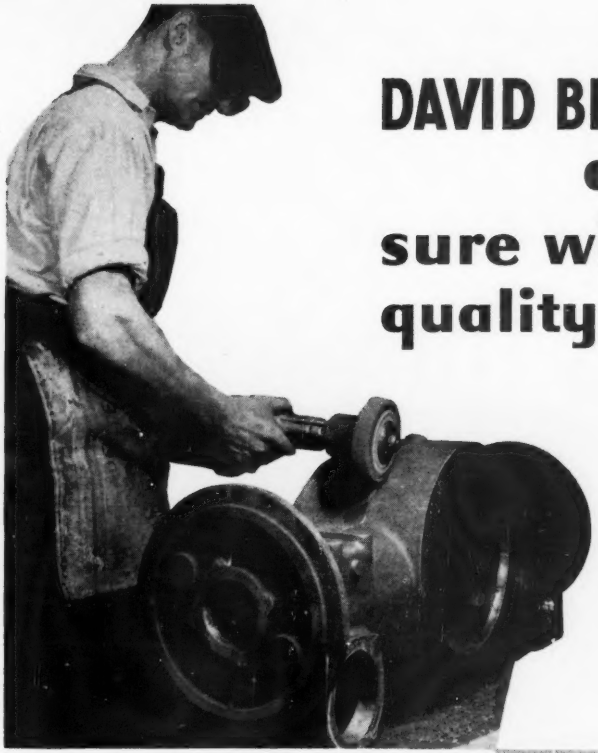

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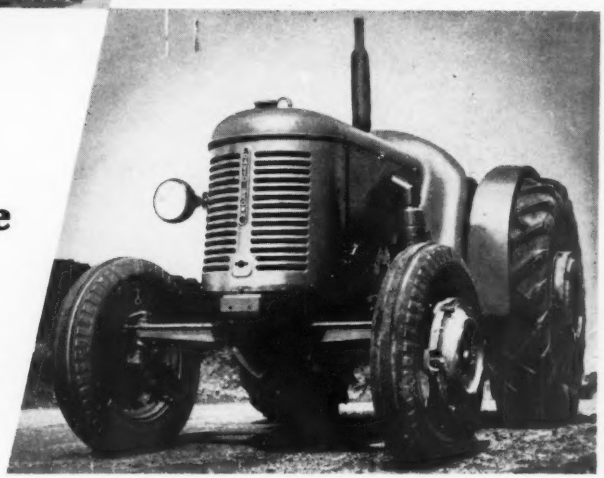

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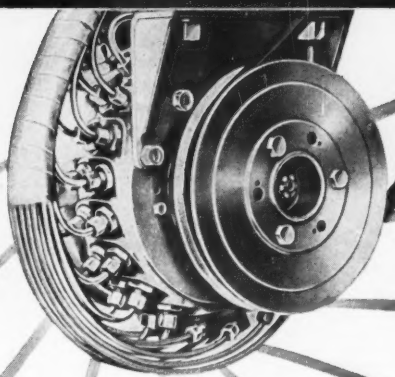

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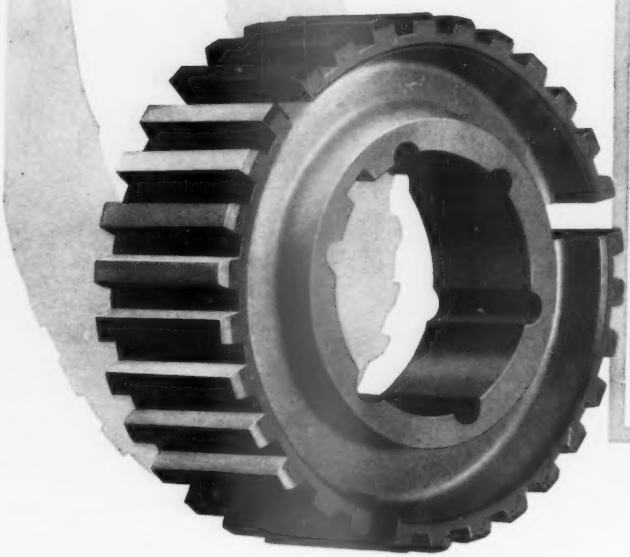
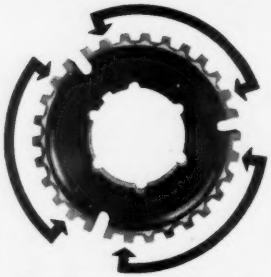
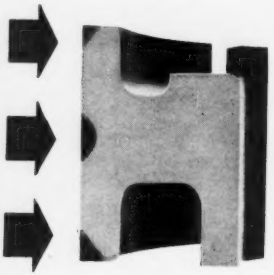




Above: View showing 3 hardened segments.

Below: Section showing 3 hardened bands.



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Production . . . . . 3500 parts per 44 hour week.

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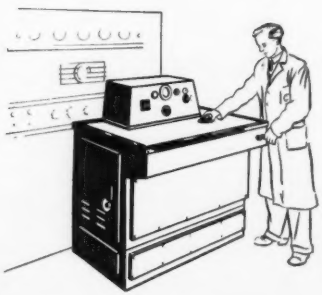

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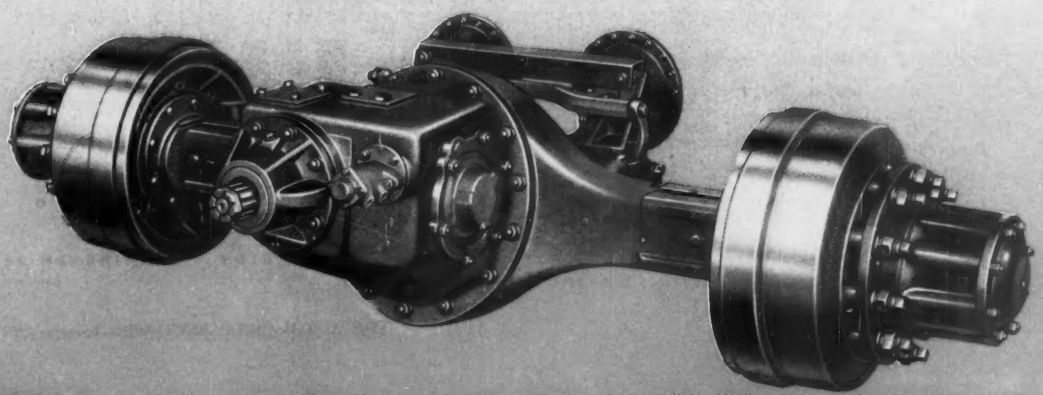

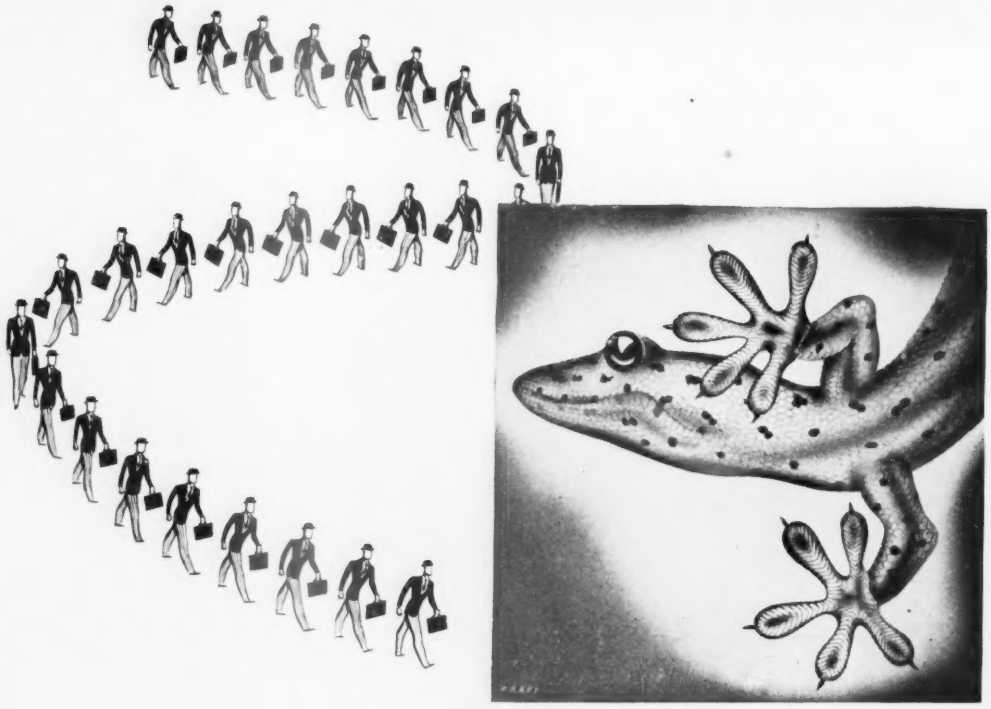

27 car manufacturers and geckonida

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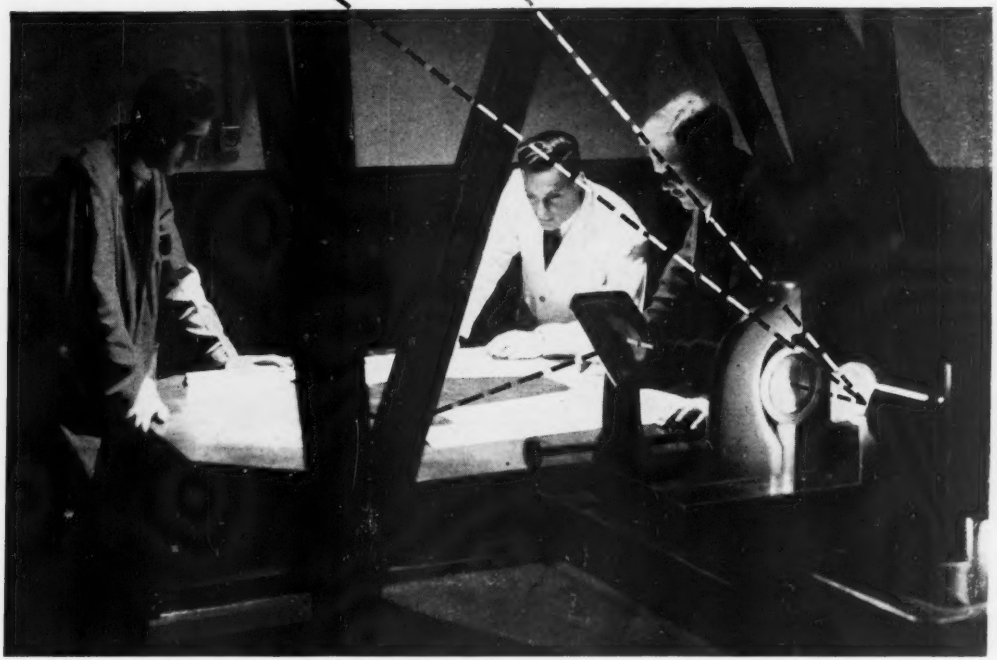

We keep our eyes peeled for inaccuracy all the way from the blank to the finished gear; and to help peel our eyes, we have an extensive range of gear testing equipment, much of which has been developed by ourselves. Other equipment, like the Taylor-Hobson projector illustrated, is brought in for examination of gear profiles, gauges, cutters, screw threads, etc.

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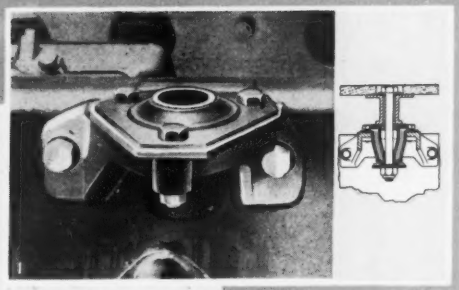

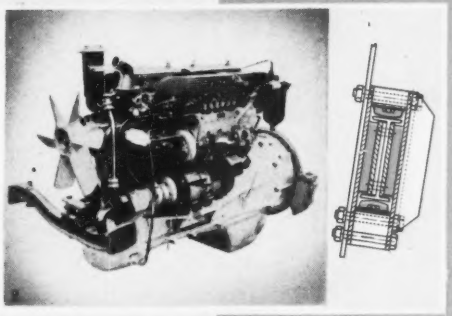

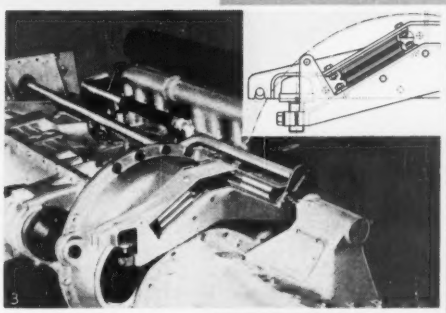

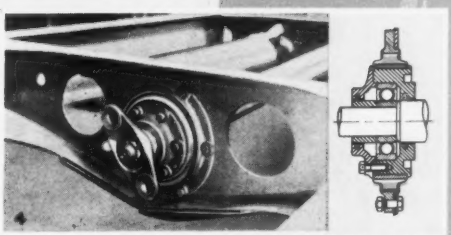

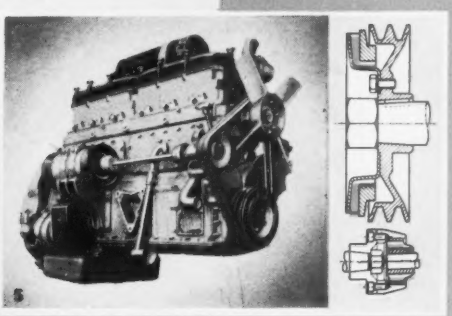

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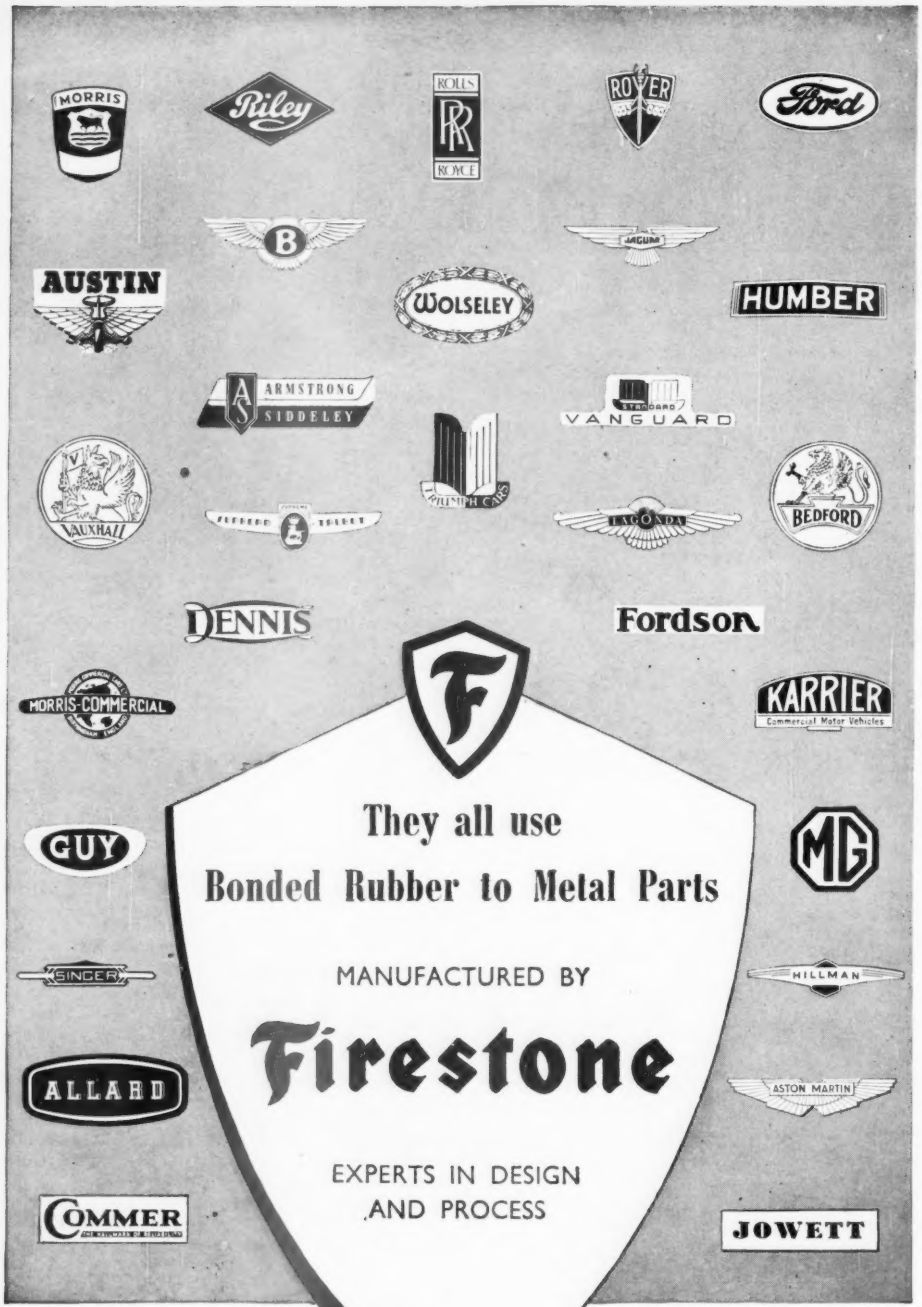

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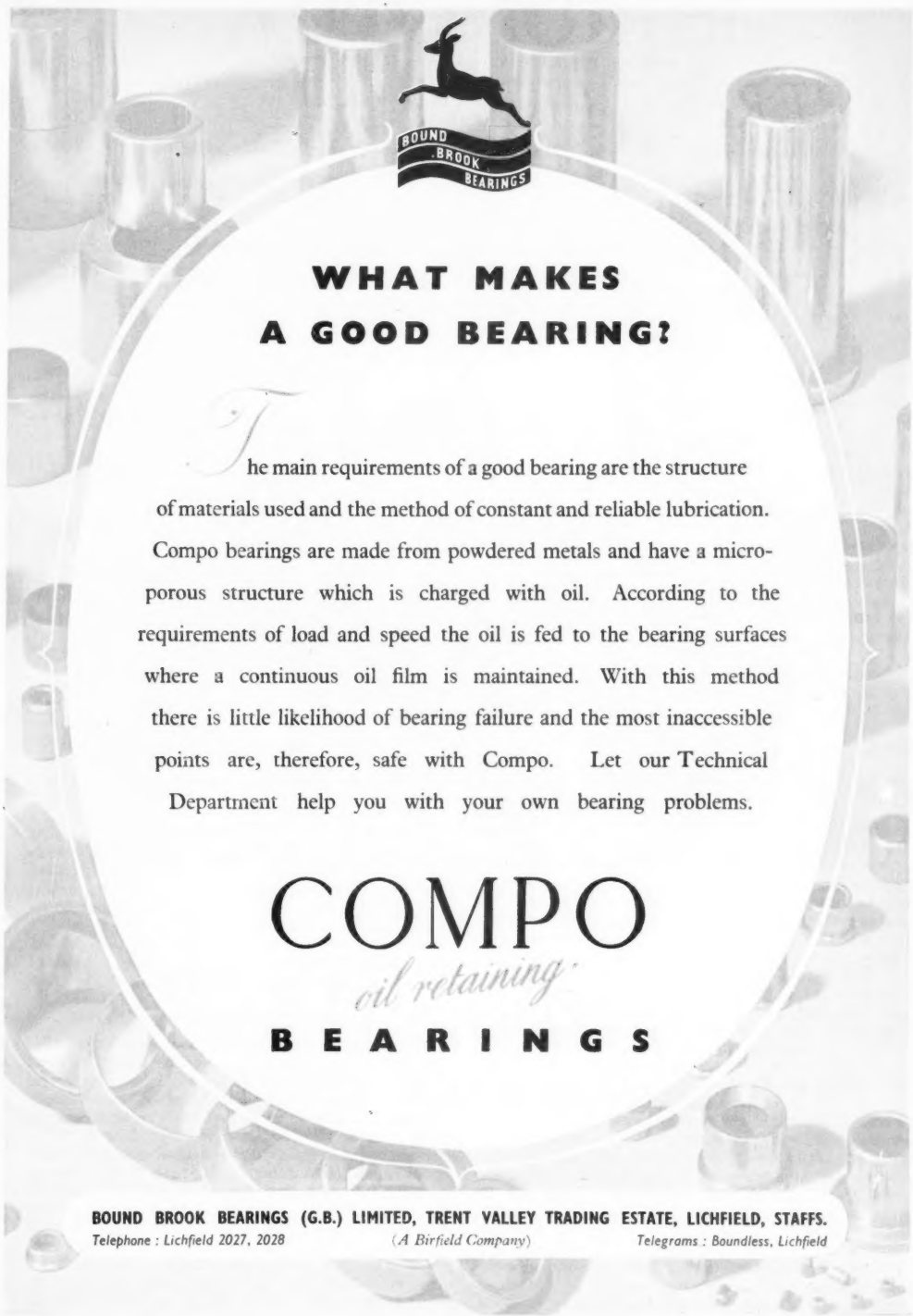

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Standardization of Components

STANDARDIZATION is a matter regarding which the automobile industry has heard much during the post-war period. It was frequently put forward by one or another spokesman of the late Government as a panacea for all the imagined ills of the industry. Moreover, it was often advanced in a manner which suggested that standardization was new and revolutionary, a method about which the industry had no previous knowledge. The fact, of course, is that the engineering industry of this country in general, and the automobile industry in particular, has for many years realized the benefits that standardization, intelligently applied, can bring.

The British Standards Institution has been in existence for fifty years and it has done a vast amount of work in many different spheres. The S.M.M.T. has had its standardization committee for some years. Standardization was also very ably advocated by the late Oliver Lucas, because of the great variation in electrical components that were manufactured by Joseph Lucas Ltd. to meet the demands of the motor industry in the pre-war and immediate post-war period. Since 1945 an appreciable measure of standardization of electrical equipment has been achieved by consultation with individual manufacturers.

There is, therefore, nothing new in standardization. Indeed, the large manufacturers have for many years followed a policy of standardization within their own concerns and as regards their own particular products.

Changed economic conditions

In the pre-war period, with its easier economic conditions, there was little need for any great degree of standardization between the various large manufacturers, except in certain special cases. Electrical equipment, as already mentioned, was one of those. The very different economic conditions of the present day, however, obviously make a greater degree of standardization not only acceptable but desirable. In 1948, at a meeting of the S.M.M.T. Production Efficiency and Standardization Committee at Earls Court on November 4th, it was decided that senior executives from the engineering and purchasing departments of the large manufacturers, known as the Big-Six, should meet for discussion at regular intervals to examine problems affecting the components used by their Companies. Between them, the Big-Six concerns are responsible for producing 87 per cent of the total output of vehicles in this country.

In this manner was formed what has since become known as the Big-Six standardization committee which, in the introduction to its first report, dated August, 1949, laid down its objective in the following words: "To promote the principles of standardization with reference to products of the Big-Six manufacturers. Firstly, to achieve improved economics, thereby lowering costs and obtaining the highest possible production and concentration upon the smallest possible variety of components and accessories. Secondly, to facilitate production methods and to promote a more efficient after-sales servicing of vehicles."

It is true, of course, that if carried to extremes standardization could result in stagnation of design, but the committee has recognized that fact, as is plain from its terms of reference set forth in the report as follows:

"To concentrate on a programme of standardization which will avoid

(a) interfering with the proper rate of technical progress
(b) interfering with a proper degree of individuality in regard to both engineering and style."

Availability

Moreover, the Big-Six do not jealously guard the results of the committee's work for the sole benefit of their constituent members, and the Standards Books which are published to record the data of standardized components are available to all manufacturers and suppliers, and to any technical body on request through the S.M.M.T.

That very valuable results have been achieved there is no doubt, and a single instance may be cited in proof of this. Thus 10 basic brake sizes have been agreed upon and as a result of this rationalization one brake manufacturer who services 136 different brake shoes will, when the scheme is in full operation, supply 10 different brake shoes to cover the full range of the 10 sizes listed.

It is obvious, of course, that standardization is, and by its very nature must be, a long-term policy. It will, therefore, be some little time before the work of the committee can come to full fruition. It is, for example, impracticable to alter a well-proved and perfectly satisfactory design merely for the sake of incorporating standardized components. To do so would seriously interfere with production, and, far from achieving economy, would probably have the opposite effect, at least for a time. When in the course of time a new design becomes necessary, as experience shows is sooner or later the case, then will be the opportunity for achieving a greater degree of standardization without concomitant economic disadvantages.

G

It is a matter for regret that the word "standardization" is apt to provoke adverse reactions in the minds of many. It has tended to connote the drastic limitation of individual choice or the imposition of a single type to the advantage of the individual or corporate producer.

Providing measures of standardization are agreed upon after consultation, compromise and adjustment, and are not exclusive in their provisions, there is no reason for an obstructive attitude. On the contrary, they should be welcomed as promoting the general advantage.

Type Reduction

The present move, in aims, methods and effects, is quite different from the drastic rationalization of the German motor industry immediately prior to the commencement of the last War. That process was conceived at political and military levels and was rapidly and ruthlessly imposed on the motor industry with all the authority and pressure of the State.

In one respect, however, the German example could be followed. To name the policy, a typical omnibus-word "Typenbereinigung" was coined. A literal interpretation would be "type-clarification" which, admittedly, does not fall easily from a British tongue. Nevertheless, it describes the policy more accurately and more aptly than does either "standardization" or "rationalization."

It is obvious that standardization on such components as electrical equipment, brakes, shock absorbers, instruments, propeller shafts, ball-bearings, parallel roller bearings, thin-wall bearings, road wheels, clutches, steering gears and substitute materials must make for a lowering of costs and the highest possible production. At the present time, when increasing competition in foreign markets is being experienced and has made inroads into last year's record export figures, it is vital to reduce production costs in every possible way in order that the automobile industry may continue to play the same important part in the economic recovery of the nation.

Kerbside Finish

THE latest addition to the nomenclature of the motor industry is the term "kerbside finish". The genesis of this new phrase, with its expressive though disturbing implications, is not really clear. If it be of transatlantic origin, which some appear to assume, we might expect that it would have come to us as "side-walk finish". Not necessarily, however, because there is the example of the kerbside parking meter, a device we believe indigenous solely to the American scene. In any event, it is the kerb rather than the side-walk upon which the new definition is based. The side-walk, or as we call it, the pavement is, after all, reserved for pedestrians. It is, in fact, on the supposition that the approach to the vehicle is no closer than the kerb, that the term "kerbside finish" owes its supposed justification.

Underlying is the idea that if the appearance of a car passes muster at a casual glance from a distance of a few feet, then that is all that is required. It is, perhaps, a matter for regret that for the overwhelmingly vast majority of mankind, a finish of this order is all that is demanded. Most of the comments on the subject, both in the industry and out of it, have been of a derogatory character. Perhaps this may be attributed to the rather unfortunate association of ideas that naturally follows the use of the word "kerb-side". There is, however, but slight reason to criticize this utilitarian approach provided the lowered standard embodies the requisite durability.

It may well be that on this whole question of superficial finish we have been, ever since the era of the horse-drawn vehicle in fact, upholding false standards. A car is an all-weather vehicle and should be finished to withstand climatic variations, much as is footwear or clothing material. To finish it in such a manner that only by incessant toil can the original appearance be maintained is hardly logical.

The crux of the matter would, therefore, seem to lie in the way the term "kerbside finish" is interpreted. Should it be made the excuse for thinner and shoddier surfaces with still less underlying protection against damp and corrosion, then it is, of course, wholly bad. If it merely means rather less sheen and fewer decorative areas of indifferent chromium plating, then no great harm will be done.

There would, however, be no very general outcry if the motor car of ordinary commercial production took on an altogether harder and more durable work-a-day finish. It is to be hoped that the new look, if and when it materializes, will be more, rather than less, durable than before. It is preferable for the car to start its working life at a lower level of shine and maintain it throughout a period of years rather than deteriorate with the rapidity that now occurs if continual washing and polishing is neglected.

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THE JAGUAR MK VII

A High-speed Touring Saloon of Outstanding Merit

WITH the introduction of the XK120 Jaguar two-seater having an entirely new twin overhead camshaft engine, it appeared inevitable and rational that the old-established pushrod engine in the Jaguar saloon models would eventually be superseded. After two years of remarkable success in motoring competitions of all types, the new engine was embodied in a new chassis known as the MK VII, carrying a redesigned body of appreciably greater capacity than the MK V saloon, its immediate predecessor.

Although outside the scope of these notes, the body has a number of unusual detail features worthy of mention. Exceptional luggage capacity has been obtained by the use of two fuel tanks mounted in the rear wings, carrying 8 gal and 9 gal respectively, each tank having an S.U. electric pump drawing through a separate fuel pipe. Tools are carried in hinged containers mounted in the front doors, thereby reducing the thickness of the boot lid, which in earlier Jaguar models housed these items. The jack and wheel brace, however, are carried in the boot, adjacent to the spare wheel which is mounted vertically along the right side.

A fresh-air heating and ventilating system is standard equipment, and among the minor body details of note is the provision of safety catches on the rear doors that prevent accidental opening from the inside. This is particularly valuable if small children are to be carried.

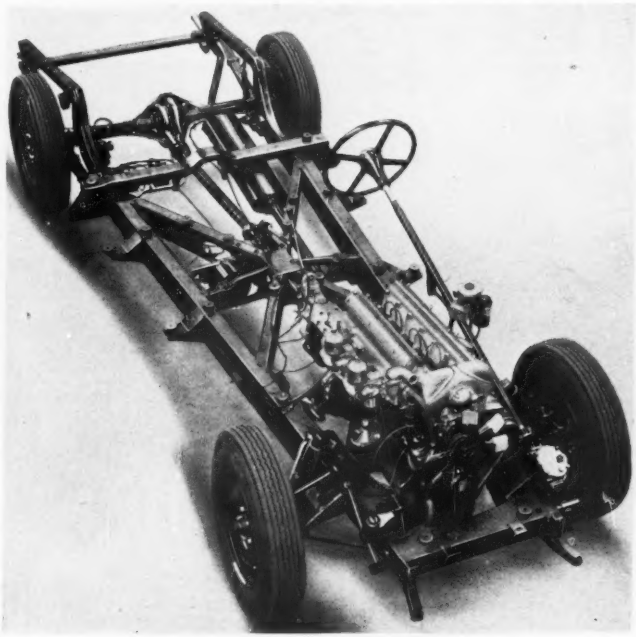
Regarding the chassis, the engine and gearbox are virtually identical with the standard units embodied in the XK 120 two-seater. The forward mounting of the engine has resulted in a modification to the sump, while an air silencer-cleaner has been added. The two-stage propeller shaft has a flexibly mounted midship bearing, and the rear axle is a hypoid bevel unit of Salisbury manufacture.

SPECIFICATION

ENGINE: Six cylinders. Bore and stroke 83 mm × 106 mm. Swept volume, 3,442 cm³. Maximum b.h.p. 160 at 5,200 r.p.m. Maximum b.m.e.p. 140 lb/in² at 2,500 r.p.m. Compression ratio 7:1 or 8:1. Seven-bearing crankshaft, twin overhead camshafts. Twin S.U. carburettors.
CLUTCH: Borg and Beck 10 in diameter. Single dry plate.
GEARBOX: Four speeds and reverse with constant load synchromesh on top, third and second. Ratios: top 1:1, third 1·367:1, second 1·982:1, first and reverse 3·375:1.
REAR AXLE: Salisbury hypoid bevel. Ratio 4·27:1 (11:47).
FRONT SUSPENSION: Independent by wishbones and torsion bars, with anti-roll bar and Girling telescopic dampers.
REAR SUSPENSION: Semi-elliptic springs with Girling piston-type dampers, type P.V. 7.
BRAKES: Girling hydraulic, self-adjusting, with two-trailing shoes at the front. Clayton Dewandre vacuum servo.
TYRES: Dunlop 6.70—16.
DIMENSIONS: Wheelbase 120 in. Track, front 56 in, rear 57½ in. Turning circle 36 ft. Ground clearance 7½ in. Weight, dry, 3,696 lb.

An X-braced frame having box-section side members forms the basic structure, to which is attached an independent front suspension assembly similar in general to that already embodied on the XK120 and MK V models. However, to permit the forward mounting of the engine the anti-roll bar has been re-positioned behind the suspension assembly, while the steering layout now incorporates double-armed levers on the drop arm and idler drop arm to bring the centre section of the three-piece track rod well forward.

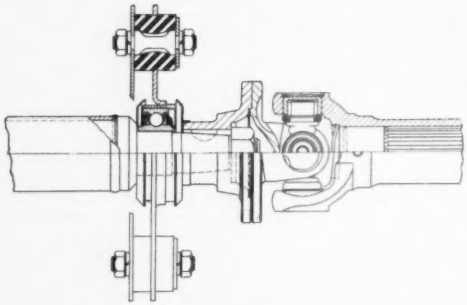
Consistent braking performances is a matter of great importance on a high speed touring car of considerable mass, and the MK VII embodies the new Girling self-adjusting, hydraulic brake having two trailing shoes at the front. This arrangement, while reducing the fade sensitivity of the brake requires servo assistance if moderate pedal pressures are to be maintained, and in this case a Clayton Dewandre vacuum servo is provided.



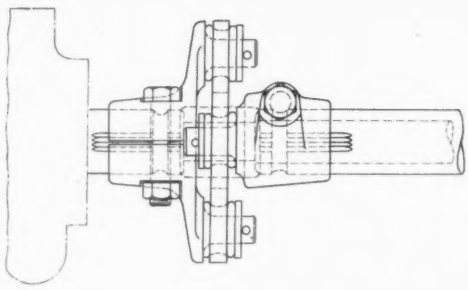
Chassis in semi-plan, showing frame cross-bracing

In recent years Jaguar cars have acquired a name, both in Britain and overseas, for remarkable value for money. The MK VII is no exception, but it should be emphasized that no concession has been made to cheapness in achieving this reputation. By any standard the MK VII represents performance, and indeed luxury, in car travel of a very high order.

Having a bore



Propeller shaft midship bearing



Flexible coupling in steering column

and stroke of 83 mm × 106 mm, giving a swept volume of 3,442 cm³, the engine develops 160 b.h.p. at 5,200 r.p.m., with a compression ratio of 8:1. The maximum b.m.e.p. is 140 lb in² at 2,500 r.p.m. An alternative compression ratio of 7:1 is available.

A detailed description of the engine and gearbox was given in *The Automobile Engineer*, for July, 1950, in connection with the XK120 model. Briefly, the cylinder block and crankcase is an integral structure of chrome cast iron, extending down to the crankshaft centre-line, with full-length water jackets surrounding the bores. The seven main-bearing housings carry steel-backed, white-metal bearings having single circumferential oil grooves with two feed holes in each half-bearing.

Of En 16, the counterbalanced crankshaft has at its front end a bonded rubber type of torsional vibration damper which also carries the fan pulley. This is pulled up on a taper by the starting handle dog, and secured by keys. Between the damper and the front crankshaft journal are the oil seal sleeve, flinger, camshaft-drive sprocket and oil pump and distributor-drive gear. End thrust is taken on the centre main bearing.

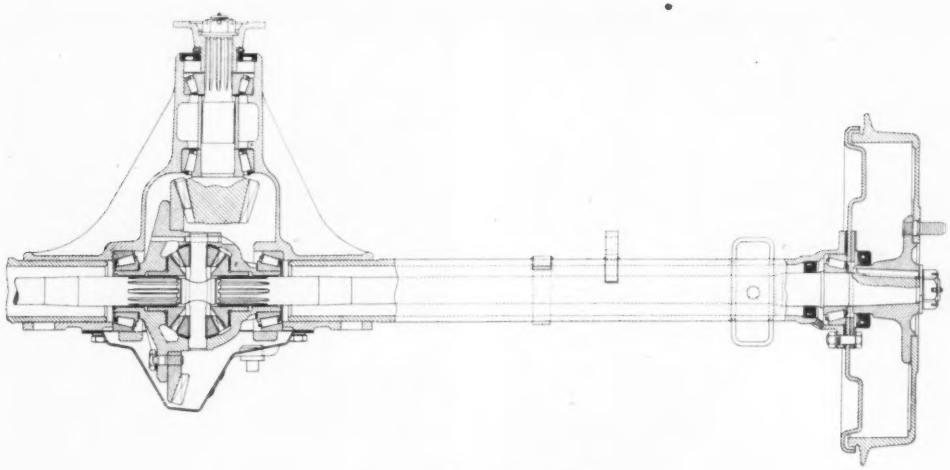
As in the case of the main bearings, the big-end bearings are of steel-backed white metal. The En 16 connecting rods have a $\frac{3}{16}$ in diameter drilling to feed oil to the small end and have a centre distance of 7·75 in. Tin-plated, aluminium alloy, solid-skirt pistons are fitted and each has two compression rings and one slotted scraper ring. The gudgeon pins are fully floating.

Carried in an aluminium alloy cylinder head, the valves are set at an included angle of 70 deg and seat on Brimol inserts in the hemispherical combustion chambers. They are operated from the twin overhead camshafts through the medium of cylindrical tappets of chilled cast iron, working in cast iron liners in the head.

Two-row roller chains drive the camshafts, the first-stage chain from the crankshaft having a blade-type tensioner and driving an intermediate sprocket. From the intermediate sprocket the second-stage chain passes around the two camshaft sprockets, between which there is an idler sprocket eccentrically mounted for adjustment. Removal and replacement of the cylinder head is carried out without disturbing the valve timing.

Included in the lubrication circuit

Three-quarter rear view of chassis



Arrangement of hypoid bevel rear axle

is a Tecalemit full-flow filter, and the aluminium alloy sump has a capacity of 22½ pt. Twin S.U. carburettors supply mixture to a six-port, water-jacketed manifold having midway a balance orifice into which the thermostatically controlled starting carburettor delivers mixture. Ignition is by coil, with centrifugal and vacuum advance mechanism.

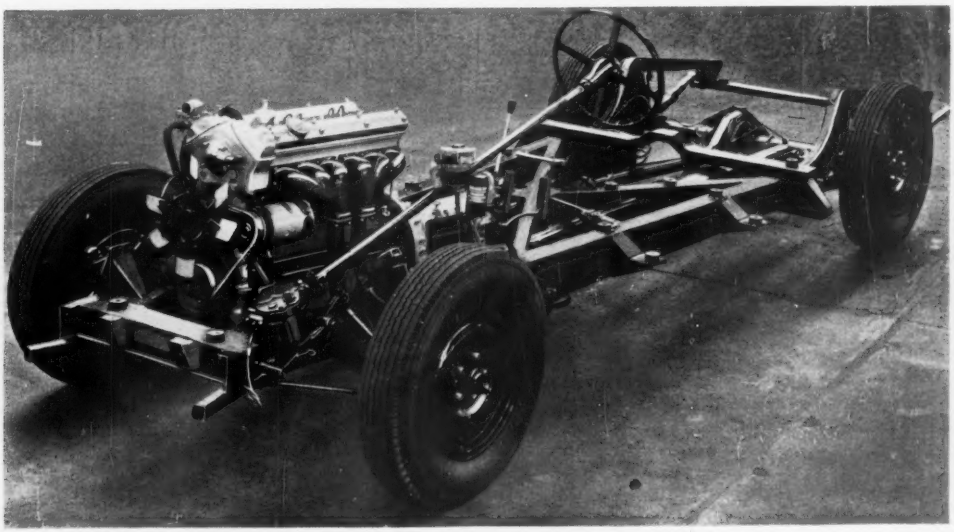
Clutch

A 10 in diameter Borg and Beck clutch is fitted. It is hydraulically operated by means of a slave cylinder mounted on the bell housing, with a master cylinder operated by the clutch pedal and mounted on a bracket attached to the frame. A separate hydraulic fluid reservoir is provided, adjacent to the brake fluid reservoir.

Gearbox

Identical in design with earlier models, the gearbox has constant-load synchromesh mechanism on top, third and second speeds, and provides the following ratios:—top 1:1, third 1·367:1, second 1·982:1, first and reverse 3·375:1. The main casing is of cast iron and carries the first motion shaft in a ball bearing. A needle-roller spigot bearing supports the front end of the mainshaft, the rear end of which is carried in a roller bearing in the back wall of the main casing. The extreme end of the shaft is supported in a ball bearing housed in a light alloy extension casing. Needle-roller bearings are also used to carry the third and second speed mainshaft gears, and the layshaft assembly.

A remote control is provided with a fairly short and stiff lever, an arrangement giving a degree of precision and



Three-quarter front view of chassis

Independent front suspension. Front view showing divided drop arm

Rear view of independent front suspension showing mounting of anti-roll bar

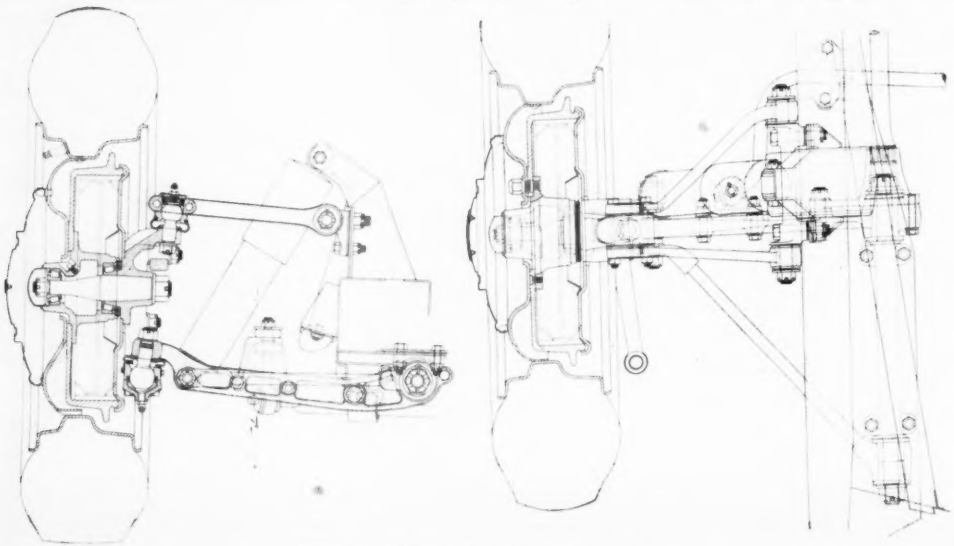
certainty in gear changing that, in some instances, is lacking in the more generally favoured steering column control.

Propeller shaft

From the gearbox a two-stage Hardy Spicer propeller shaft with a flexibly mounted midship bearing transmits the drive to the rear axle. As shown in the illustration, the midship ball bearing housing is supported on a coil spring that carries the weight of the shaft. Vertical and lateral location is by means of two rubber bushes of 38-33 durometer hardness, held on necked pegs secured to the frame. The $\frac{21}{32}$ in bore of the rubbers seats on lands at the ends of the pegs with an interference fit of $\frac{1}{32}$ in. The $1\frac{17}{32}$ in outer diameter of the rubbers provides an interference of $\frac{1}{32}$ in in the flange of the bearing housing. It will be evident that the assembly provides a measure of controlled flexibility, dependent upon the stiffness of the rubber bushes. This is selected so that the natural frequency of the mounting is in resonance with propeller shaft disturbances at a road speed of about 10 m.p.h.

Rear axle

Of semi-floating type, the Salisbury hypoid bevel rear axle, providing a pinion offset of $1\frac{3}{4}$ in, has a ratio of 4·27:1 (11:47). With the tyres specified the road speed in top gear is 19·35

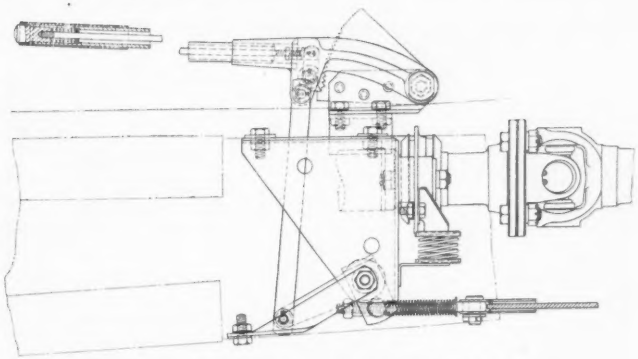


Elevation and plan of independent front suspension

m.p.h. per 1,000 r.p.m. of the engine crankshaft.

The general layout of this type of axle, with a malleable iron centre housing into which are pressed the side tubes, is now well known. One of the major advantages of this construction is the lateral support provided for the outer races of the differential bearings, which butt against continuous surfaces in the centre housing.

Taper-roller bearings are fitted throughout, with shim adjustment. The hub bearings are pressed on to the axle shafts and are located in the ends of the axle tubes. Outward thrust on a road wheel is taken by the adjacent bearing, while inward thrust is transmitted to the bearing in the opposite hub via a spacer against which the inner ends of the shafts abut. The axle shafts are $1\frac{7}{64}$ in minimum diameter and the hubs are secured to the shafts by a 1-in-12 taper and key.



Arrangement of hand brake over spring-supported midship bearing

Suspension

Following closely the layout of the XK120, the independent front suspension embodies wishbones and torsion bars giving an effective static deflection at the wheel of 8·75 in. The stub axle carrier, as before, articulates on ball joints, but has been redesigned to provide a swivel pin angle of 8 deg instead of 5 deg, thereby reducing the offset resulting from the larger section tyre. Zero castor angle is provided, and the camber angle is 1 deg.

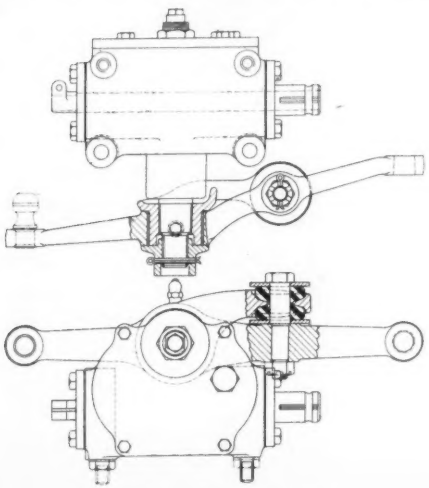
On account of the forward mounting of the engine in relation to the suspension assemblies, the anti-roll bar has been re-positioned behind the suspension and passes through rubber-bushed brackets on the underside of the frame side members. Girling direct-acting dampers are embodied.

Semi-elliptic laminated springs at the rear provide an effective static deflection of 6 in and are controlled by Girling P.V. 7 piston-type dampers. An interesting detail is the downward-turned eye at the rear end.

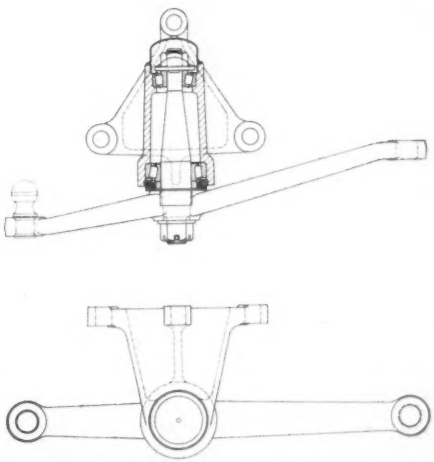
Steering layout

Forward mounting of the engine has also necessitated a new track rod layout. Although a three-piece track rod assembly is retained, the drop arm and idler spindles now carry double-armed levers of considerable length, namely 12·625 in (5 in forward of the spindles and 7·625 in rearward). The forward ends are linked by the adjustable centre tie rod $31\frac{3}{16}$ in in between ball centres, while the rear ends connect with the forward-projecting steering arms on the stub axles by the fixed tie rods $8\frac{11}{16}$ in between centres.

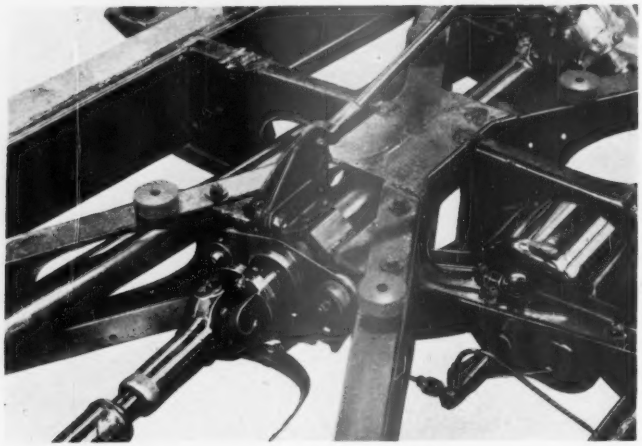
Although this layout might be considered to have appreciable flexibility, the transmission of shock to the steering wheel is still further reduced by a rubber damper in the drop arm, shown in the illustration. On the serrated end of the drop arm spindle is mounted a lever 3·5 in long terminating in a housing for rubber "biscuits". The steering lever pivots on the boss of the drop arm lever, where a Vandervell, graphite-impregnated, split bush is provided. A stepped bolt locating in the rearward extension of the lever passes through the rubbers and drop arm lever, a washer under the bolt head compressing the rubbers to the required degree. Shocks transmitted from the road wheel are thus reduced by the relative movement between steering lever and drop arm lever, cushioned by the rubber inserts.



Arrangement of drop arm damper



Idler steering arm assembly



Frame centre section showing propeller shaft, midship bearing, hand brake and servo cylinder

The steering mechanism itself is of the Burman recirculating-ball type, and the idler spindle on the opposite side is carried in taper-roller bearings. At the lower end of the steering column is a flexible coupling consisting of a $\frac{1}{4}$ in thick plate in which are housed four rubber bushes. The pins on the forks carried on the steering column and steering box shaft respectively engage with the bores of the bushes. A rubber sealing sleeve bridges the gap between the column end and shaft.

At the upper end the steering column rotates in a Vandervell, graphite-loaded, bronze bush carried in a steel housing which is itself mounted in a thick rubber bush. The column is adjustable for length.

Frame

In general, the frame members are of 14 S.W.G. material; 10 S.W.G. being used for such items as body mounting brackets. The side members are box section, the minor channels being deflected inwards towards the front end adjacent to the suspension mounting brackets. This front part of the frame therefore comprises four channel-section members, with a closing plate on the underside of the narrow triangle formed between the diverging side members. A substantial box-section cross member closes the front end.

Bracing the centre of the structure is a channel-section X-frame with channel-section transverse bracing members between the central junction and the side members. Through the junction passes the first stage of the propeller shaft; the midship bearing being supported on a bracket bolted in the apex of the junction. On the right-hand side is an additional supporting channel for the servo cylinder.

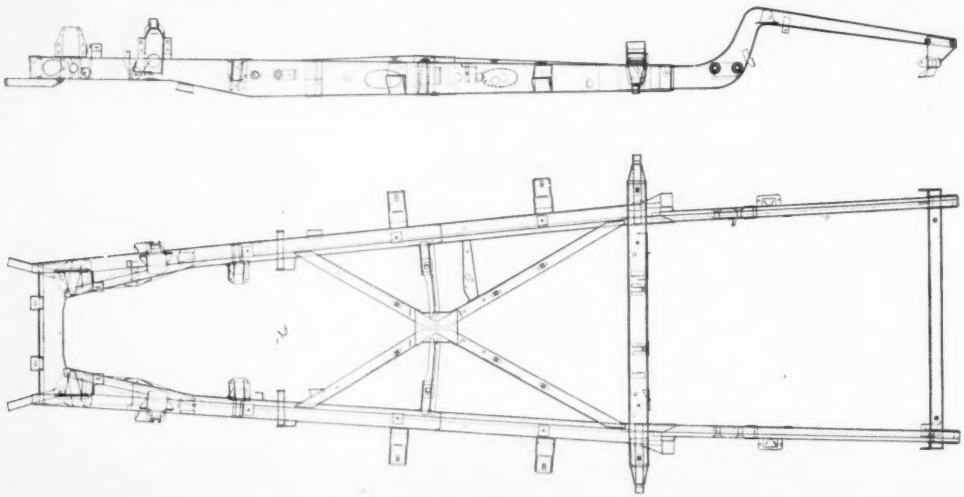
A box-section arched member spans the frame just forward of the rear spring anchorage, at which point also the separate, upswept, rear sections of the side members are welded on to the main side members. A light, box-section, cross member closes the rear end of the frame; welded to the downward extending shackle brackets.

Brakes

Two-trailing shoe Girling brakes are fitted at the front, and leading-and-trailing shoe brakes at the rear. The front brakes are automatically adjusted, and the whole system is operated hydraulically through the medium of the Clayton Dewandre vacuum servo cylinder deriving vacuum from a tapping at the rear of the induction manifold. With this arrangement no reservoir is fitted, nor does the system operate on the suspended vacuum principle. The slave cylinder is mounted directly on the rear face of the vacuum cylinder. A total braking surface of 207 in² is provided.

Dimensions

Wheelbase, 120 in; track, front 56 in, rear 57½ in; ground clearance 7½ in; turning circle 36 ft; weight, dry 3,696 lb. Tyres, Dunlop 6·70-16.



Layout of frame

TOUGHENED GLASS

A Survey of Properties, Manufacture and Application

By H. D. Hendrick

MODERN industrial requirements for materials having certain specific characteristics whilst withstanding very severe service conditions, have resulted in the use of glass for a wide range of duties. Toughened or "Armoured" glass has, in recent years, been adapted to uses in industries ranging from textiles to steelworks. The most familiar use of toughened glass is as safety glass screens or windows in passenger vehicles. To appreciate the potentialities of toughened glass, it is necessary to consider the main characteristics and properties. This article deals with the salient features and describes in broad outline the manufacturing process used to obtain the special qualities.

The process

A suggested definition for Toughened Glass is "A glass which has been subjected to heat treatment whereby high compression stresses are set up in the surface zones and correspondingly high tensile stresses in the centre".

Glass is a brittle substance because of the considerable difference between the compressive strength, which is comparatively high and the tensile strength which is low. Breakages are nearly always due to shock or applied loads which overstress the surface tension. The function of toughening is to establish permanently in the glass, a predetermined arrangement of stresses of large magnitude, the most important characteristic of which is that the stresses in the surfaces are wholly compressive. It is possible by this deliberate prestressing to set up surface compressive stresses four times as great as the breaking tensile stress.

An interesting experiment with which some readers may be familiar, is the production of "Rupert's Drops". These are drops of molten glass which are allowed to drip into water. A tear shaped particle results, the head of which has considerable strength and will withstand a heavy blow. The tail, however, is extremely fragile and is easily broken off. When this is done, the whole drop disintegrates entirely into a finely powdered glass. Upon examination it will be found that the minute particles are each a small granule. Further, it is noticeable that the granules have not sharp edges or points and that the powdered drop may be crumpled or rubbed in the hands. This experiment is a form of toughening, employing the characteristics of thermal expansion and contraction, the basis of all glass toughening procedures.

Controlled heating and quenching of glass which is described later in this article, is similar in operation to the quench hardening of carbon steels. The change effected in the material however, is totally dissimilar as glass is amorphous, non-crystalline and has no "structure" in the metallurgical sense. The effect of glass toughening is purely physical and heat treatment at temperatures below those in the range of the glass plastic condition cannot, in any way, alter this structureless condition.

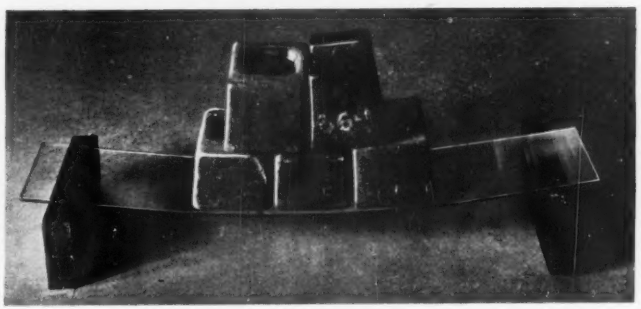


Fig. 1. A piece of "Armourplate" glass 45 in × 10 in × ¼ in thick as a simply supported beam, bearing a load of 5-56 lb weights

This physical change is dependent upon the coefficient of thermal expansion and contraction of the glass, together with its ability to flow or yield readily by small amounts when in the plastic range of temperature, much below the molten temperature. An essential feature of the process is that the whole article must be uniformly treated, both in heating and quenching, to avoid rupture by thermal strain.

The glass plate or piece is subjected to controlled heat until it reaches the commencement of the plastic range. It is then removed from the heat as quickly as possible and rapidly cooled, uniformly, over the whole of the surface. It will be apparent that at the commencement of this chilling or quenching, the surface of the glass cools and sets rigid. As glass is a poor conductor of heat there is a comparatively long time lag before the heat in the interior is dissipated. The inside is therefore, still sufficiently hot to be plastic whilst the surface solidifies; the plastic section follows the continued contraction of the glass as it cools. Throughout the chilling process, there is a difference in temperature between the surface and the interior. In the initial stages the plasticity allows the inner glass to yield to the contractions of the solidifying outer skin so that whilst there is still a big difference in temperature, there is not, as yet, any strain between inside and outside.

Eventually however, all sections must arrive at the same temperature. The outer skin cools to atmospheric temperature and ceases contracting, whilst the inside tries to continue contracting as the temperature lowers through the thickness of the glass. As it is, and must remain, firmly attached to the outer skin, this contraction is resisted so that the inside is in a state of tension. Glass, like all plastic materials, is deformed elastically by the application of a load. The inner tension therefore, loads the surface with a balancing compression.

Properties

When the toughening process is uniformly controlled it is possible to produce surface compressive stresses approximately four times greater than the tensile stress which would be sufficient to cause fracture in an ordinary annealed glass. An example of the properties of toughened glass is given by "Armourplate".* This has surface compressive stresses in the order of 20,000 lb/in². The surface tensile strength of the glass itself is about 5,000 lb/in². It follows therefore, that it is necessary to produce an apparent surface tensile stress for example, by bending, of a load about 25,000 lb/in² before final rupture. Normally the allowable working stress of a good quality toughened glass is four to five times that of an annealed plate glass, namely, from 6,000 to 6,500 lb/in². Under favourable conditions the allowable stress may be much higher.

Toughened glass has considerably

*Registered trade mark of Pilkington Bros. Ltd.

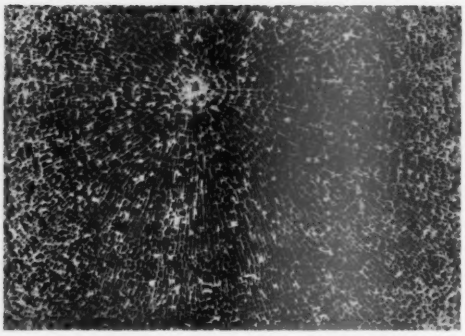


Fig. 2. "Armourplate" glass after fracture during the ball test

greater flexibility than annealed glass of the same thickness. The modulus of elasticity of the glass is not altered by toughening and it is possible to load the glass in the form of a simply supported beam until it shows four or five times as much deflection as that possible in ordinary annealed plate glass. A typical example of this is given in Fig. 1.

Shock loads

Considerable shock loads are taken by toughened glass; a typical demonstration is to drop a steel ball weighing 1 lb 11 oz, from a height of 8 ft or more upon a plate which is supported upon two sides only, as shown in Fig. 1. (Under similar conditions a drop of 15 in to 18 in is sufficient to break ordinary annealed glass of the same size and thickness.)

It will withstand local heat and thermal shock several times as severe as that which would shatter ordinary glass. For example, it may be heated to 250 deg C, and then quenched with cold water. In addition, toughened glass will withstand high stresses through a wide temperature range; One test was from minus 70 deg C, up to plus 295 deg C, without any deterioration in strength. This resistance is due to the fact that it is difficult to give any thermal treatment which will reverse the pre-stressed condition resulting from the toughening process.

If toughened glass is heated in service to a sufficiently high temperature, it will again approach the plastic condition. The existing deliberately high stresses will then be relieved by a mutual yielding of the tensile and compressive zones, reverting to the untoughened condition. The temperature above which this occurs noticeably is a few degrees below 300 deg C, and the glass usually breaks on cooling.

Safety features

Toughened glass retains the feature of ordinary glass by being unable to survive beyond its elastic limit; unlike metals it is not able to delay fracture by deformation. The internal store of elastic strain energy does however, give a "safety fracture" when the glass is broken. Failure at any point in a piece releases this strain energy by breaking down the stress arrangement, the energy then expending itself into shattering the whole piece into remarkably small granules. The granules do not readily fall apart, as shown in Fig. 2, and there is an absence of sharp edges or splinters on the small pieces.

This feature has resulted in the adoption of certain types of toughened glass as a safety glass, covered by British Standard Specification No. 857. One specified condition is that there are a minimum number of granules, dependent on the thickness, to a unit area when the glass is disintegrated. A characteristic example of this is shown by Fig. 3.

Cutting or working

Toughened glass must not be cut, ground, drilled or otherwise worked. Any attempt at this after toughening will most certainly release the strain energy, causing immediate disintegration. The glass must be cut to shape, bevelled, drilled, etc., before the toughening process. Toughened glass cannot be supplied with the edges as cut, the normal edges being:—

i arrissed or smooth arrissed
ii ground
iii round polished
iv flat polished

Glass bevels of any width up to half-inch can be toughened, but the thickness remaining at the edge must be not less than one-eighth inch.

For drilling, there are certain limits to the size of hole which can successfully be produced. These limits are usually accepted as given in the table at foot of the page.

Care must also be taken with regard to the position of holes. Where a hole is drilled near the edge of a plate, the width left between the edge of the hole and the edge of the glass must be at least equal to the thickness of the plate. A hole drilled near a corner should not be nearer than six times the thickness. Where two or more holes are required close together, the distance between must not be less than four times the plate thickness. Square corners for holes or notches are not permissible and the minimum radius of corners must be equal to the thickness of the plate.

It will be noticed that most pieces of toughened glass have small indentations or "dimples" along one edge. These are grip marks made by supporting toggles, which are described later in this article. The dimples are usually kept as close as possible to the edge and are covered by the supporting frame when the glass is in use.

Manufacturing equipment

It has been emphasized earlier in this article that uniformity of treatment is most necessary in the toughening process. Close control, preferably automatically operated, is therefore essential. The most convenient method of supporting the glass is to hang the plate vertically to ensure that both sides are equally treated. Toggles for supporting the glass are a small but very important part of the equipment. Often the cause of faults or failures may be traced to the method of suspension. The toggles are fitted with a small projection to grip the glass. Various types of toggles are in use, the simplest being in the form of an inverted "U" type spring, fitted with a collar which may be moved downwards

Glass thickness	Minimum hole diameter	Maximum hole diameter
2·6 to 3·2 mm / 3·2 to 4·2 mm	$\frac{1}{8}$ in	
$\frac{3}{16}$ in to $\frac{3}{8}$ in	$\frac{3}{16}$ in	Not greater than $\frac{1}{4}$ of the width of the plate
$\frac{3}{8}$ in to $\frac{1}{2}$ in	$\frac{1}{4}$ in	
Over $\frac{1}{2}$ in	Not less than half the thickness	

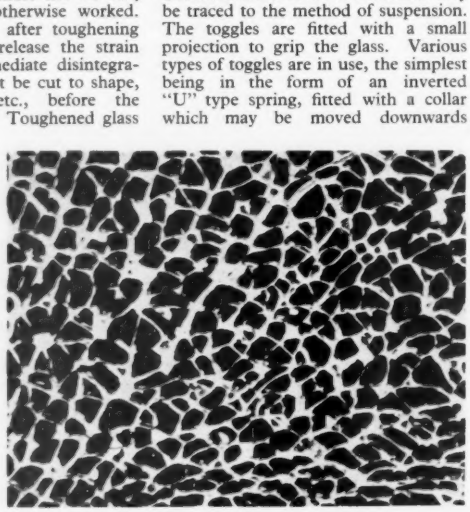


Fig. 3. An actual size reproduction of the characteristic disintegration of toughened glass into granules when broken

forcing the ends of the "U" together.

A more complicated toggle but one which has proved highly successful, is the lattice tong or scissors type, where the weight of the glass causes the projections to tightly grip onto the glass. Whatever the method of gripping, it is essential that the toggles are as free as possible where supported on the toggle bar or carriage as there will be differences in the thermal contraction and expansion of this bar and the glass. Any resistance to this expansion or contraction will cause the glass plate to bend when in the plastic condition. Where the plate is bent or curved before chilling, the toggles must be free to swivel. In the design of toggles, provision should be made for as much adjustment as possible and a nickel chromium steel should be used to minimize oxidation when subjected to heat.

Furnace

The heating or "cooking" of the glass is a most important part of the process and furnaces are designed to meet the special and rather stringent requirements. Electric furnaces are most successful owing to their inherent cleanliness and complete absence of products of combustion, effective automatic control, and above all, the rapid and uniform heat transfer possible from resistance heating elements.

Most toughening furnaces are of the batch or intermittent type, that is, one batch of glass is charged, heated and discharged before the next enters the furnace. For large scale production there are long horizontal furnaces through which the glass progresses slowly passing through regulated heating zones to the discharge end from which they are rapidly transferred to the chillers. The intermittent furnaces may be divided into four types :—

(a) Horizontal double ended, where the glass enters at one end and is discharged at the other.
(b) Horizontal single ended, the glass being charged and discharged from the same end.
(c) Vertical top loading, the glass being lowered into the top of the furnace and lifted out through the same opening. This type of furnace is usually in a pit.
(d) Vertical bottom loading, where the furnace is supported above the chillers and the operation is the reverse of that described in (b).

On types (c) and (d), the loading and unloading is usually on the same level as the chiller gear, a horizontal track being provided at one or both sides to take a trolley carrying the toggle bar. The furnaces are basically, two radiant panels one on each side of the glass, the panels being controlled within close temperature limits across the area. It is also most important that the temperature and heat transmitted from each side is equal. In electric furnaces, heating elements are usually either of nickel chromium wire or strip, the former being in the form of a coil, the latter arranged in a sinuous pattern. Coiled wire heating elements are housed in refractory bricks forming the sidewalls, these preferably having open grooves or slots to contain the elements and permit free dissipation of heat to the chamber. Strip elements are hung upon special supports fixed to the furnace wall. On some furnaces, it is necessary to have additional or auxiliary heating elements in the base of the chamber. To ensure a long operational life of the heating elements, these are often fed from a low voltage supply enabling the use of a heavy gauge wire or strip. Considerable care must be taken in the design of elements as a high rate of heat transfer is essential to bring the glass into the plastic range so quickly that the plate may be removed before deformation or "dropping" takes place. The dissipation of heat from the surface of the element must be sufficiently high to give this rapid transfer but at the same time care must be taken that the element is not over-rated, reducing the service life.

Fig. 4. A glass screen on a charging machine for an electric furnace at the Works of the English Steel Corporation Ltd.

To ensure the necessary temperature uniformity, the elements are graded from top to bottom of the sidewalls to give a predetermined input. Heating elements are also "zoned" or divided into banks, each independent and automatically controlled. This arrangement provides for fluctuating conditions such as momentary heat losses through an open furnace door and differing shapes and sizes of glass. In order to give a rapid recovery of furnace chamber temperature when a fresh plate of glass has been inserted, some form of voltage regulation is an advantage The most satisfactory method is a proportionate or "floating" controller which automatically operates a variable voltage regulator transformer. This increases the voltage to heating elements when the temperature has fallen, and consequently a fine degree of control is obtained.

Furnace linings are usually of a hot face insulation which has a low density, thereby having a reduced heat content. This is advantageous for quick heating up as less heat is required to soak the furnace brickwork to stable conditions. On the other hand a dense refractory lining which absorbs considerable heat acts as a reservoir to maintain a thermal balance in the furnace chamber. In all furnaces the lining is backed by an insulation, usually calcined diatomaceous earth, to minimize heat losses by radiation and convection from the outer face.

Chiller gear

Uniformity in chilling or quenching is just as important as in the heating process and excessive variation in the application of the cooling medium across the area of the glass will shatter the plate. Various methods have been employed and discarded, including oil quench tanks into which the glass is quickly immersed to panels of carbon dioxide refrigeration. Forced convection by the use of cold air to remove the heat from the hot glass is now universally adopted. Continental manufacturers generally use high pressure fans blowing directly through a funnel-shaped distributor on to the face of the glass. In this country and in America, compressed air is usually preferred, especially in the manufacture of safety glass for use in passenger vehicles.

Where compressed air is used, the chiller frame is usually in the form of a vertical battery of small bore pressure tubes on each side of the glass, fed from two main headers. The small bore tubes have, at a predetermined pitch, very small holes drilled so that numerous jets of air at a high velocity are directed simultaneously on both sides of the glass. To provide the necessary turbulence and ensure that all the surface is equally treated, the chiller frames are agitated during blowing. There are various methods of agitation, some of which are patented, the most commonly employed being

as follows :—

(1) Oscillating type, where the vertical frames are rotated in the vertical plane by the use of an eccentric or cranked drive. On this type each jet travels rapidly in a complete circle.
(2) Linear type, the frames being made to move to and fro in either the vertical or horizontal plane.
(3) Stationary frame type, where the individual pressure tubes are moved through an arc in either vertical or horizontal plane, or alternatively, rotating jets are employed.

It will be noted that any of these methods provides a distribution of the air jets over the glass surface, the arrangement described under (1) being the most effective. The time taken between removing the glass from the furnace, to the initial quench must be kept to a minimum, preferably of the order of five seconds. To give instantaneous operation of the chillers when the glass is in position, a quick action solenoid valve is usually incorporated in the air line so that the compressed air is automatically and immediately released to the chiller frame when the glass enters this section. A similar operation is used to switch on the drive gear for agitating the chiller frames.

Air compressors

The compressor normally used for producing cooling air is the two stage double acting type which has one low pressure cylinder in which the air is compressed to an intermediate pressure, passing through an intercooler before entering the final high pressure cylinder. An after cooler is usually provided to reduce the delivered air temperature and thus free it from the oil and moisture which would be detrimental to the process.

On intermittent plants the chiller operates, usually between eight and sixteen times per hour ; for continuous plants the number of operations are correspondingly increased. It will be evident that the requirements for air demand a large capacity for short periods ; it follows therefore that the compressors must be large enough to provide sufficient air required in a specified time and receiver capacity sufficient to store this air when the chiller is not in operation. Sufficient air receiver capacity is most important. It is essential also to ensure that air mains are large enough to give the rate of air flow required during the blowing period. To reduce the amount of large diameter air mains, it is preferable that the compressor is placed as near as possible to the chiller frame, In addition it is usually necessary to fit an oil and moisture separator close to the chiller frame to take care of any condensation which may have taken place in the air mains after the air has left the receivers.

Present and potential applications of toughened glass

The high resistance of toughened glass to impact and thermal shock has opened up many new fields of application where ordinary annealed glass could not be used. Safety glass to British Standard Specification No. 857 is used extensively in passenger vehicles including public transport. The Board of Trade now permit "armourplate" glass for port holes, etc., in certain parts of ships where normally, very thick polished plate glass was necessary to withstand the terrific pressure of wave shock.

Resistance to thermal shock has extended the field to commercial and domestic applications. A recent innovation is the fitting of toughened glass doors to gas and electric ovens so that the cooking or heat treatment process may be observed without wasting heat by opening the doors. This resistance to heat is also useful for table or counter tops in restaurants and canteens, or for drying tables in chemical plants. Toughened glass is extensively used to counter thermal shock in high powered floodlighting and similar applications.

In addition, toughened glass in various forms, has been used with considerable success, for example, as glasses for meters, machinery guards, catering cooking ranges, hospital trolleys, miner's cap lamps, roof and pavement lenses and doors. Many other possible applications will be evident to designers requiring a robust material offering a clear vision or permitting the filtration of light.

One typical industrial application is shown in Fig. 4. This shows a glass screen on a charging machine for an electric furnace at the works of The English Steel Corporation Ltd., Sheffield. The screen gives the driver complete protection against splashes of molten metal and at the same time gives a clear, undistorted view of the furnace.

In conclusion the author wishes to express grateful acknowledgment to Pilkington Bros. Ltd., for much of the information contained in this article and for the use of photographs, to James Royce Electric Furnaces Ltd., and Broom & Wade, Ltd., for technical details on manufacturing equipment.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during January :—

BIRMINGHAM CENTRE

Tuesday, 22nd January, 6.45 p.m. General Meeting in the James Watt Memorial Institute, York House, Great Charles Street. Paper: "Shock Absorbers," by J. W. Kinchin and C. R. Stock.

DERBY CENTRE

Monday, 21st January, 6.45 p.m. Address by the Chairman of the Automobile Division, Mr. C. B. Dicksee, M.I.Mech.E., entitled "Experiences during Twenty Years of Oil Engine Development."

LUTON CENTRE

Monday, 14th January, 7.30 p.m., in the Town Hall Assembly Room. Paper: "Trailers and Semi-Trailers," by A. Marenbon.

Tuesday, 22nd January, 7.30 p.m. Extra General Meeting in the Town Hall Assembly Room. Address by the President, A. C. Hartley, C.B.E., B.Sc.(Eng.), M.I.Mech.E.

NORTH-EASTERN CENTRE

Wednesday, 16th January, 7.30 p.m. General Meeting in the University Chemistry Lecture Theatre, Leeds. Paper: "The Design and Development of Very Large Road Haulage Vehicles," by C. E. Burton, M.I.Mech.E.

NORTH-WESTERN CENTRE

Wednesday, 23rd January, 7.15 p.m. General Meeting in the Walker Engineering Laboratories, The University, Liverpool. Paper: "Trailers and Semi-Trailers," by A. Marenbon.

SCOTTISH CENTRE

Monday, 21st January, 7.30 p.m. General Meeting in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "Trailers and Semi-Trailers," by A. Marenbon.

WESTERN CENTRE

Thursday, 31st January, 6.45 p.m. General Meeting in the Royal Hotel, Bristol. Paper: "Independent Rear Suspension," by Donald Bastow, B.Sc.(Lond.), M.I.Mech.E., (Member of A.D. Council).

The following meetings will be held during February :—

LONDON

Tuesday, 12th February, General Meeting at Storey's Gate, St. James's Park, S.W.1. Paper: "Noise in Private Cars," by J. R. Bristow, B.Sc., Ph.D., A.M.I.Mech.E.

COVENTRY CENTRE

Tuesday, 5th February, 7.15 p.m. General Meeting in the Craven Arms Hotel, High Street. Informal Discussion.

DERBY CENTRE

Thursday, 7th February, 6.15 p.m. General Meeting in association with the East Midlands Branch in the Rolls-Royce Welfare Hall. Repetition of Thomas Hawksley Lecture by H. Roxbee Cox, Ph.D., B.Sc.(Eng.), M.I.Mech.E., entitled "Some Fuel and Power Projects."

LUTON CENTRE

Monday, 11th February, 7.15 p.m. Annual Meeting followed by General Meeting in the Town Hall Assembly Room. Talk by Dr. D. B. Foster on the subject "Carbon as an Engineering Material."

INDEPENDENT REAR SUSPENSION*

Potential Advantages and Requirements

INDEPENDENT front suspension serves mainly to permit softer front springing without incurring the corresponding disorders. Independent rear suspension (I.R.S.) has quite different potential advantages. They are:—

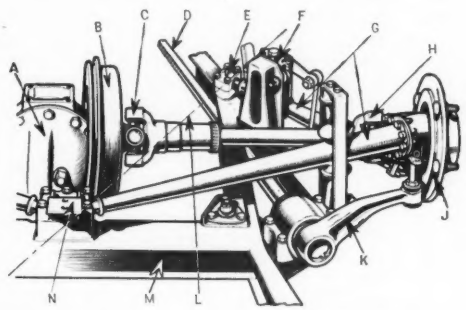
(1) Reduction in unsprung weight and, hence, an improved ride. Better road-holding when cornering or braking.
(2) Absence of propeller-shaft torque effect and therefore improved traction in extreme conditions, as with low gear and full throttle. Elimination of the difference in handling as between left-hand and right-hand corners taken under power.
(3) Elimination of various hops, shakes and tramps of the rear axle, and reduction of others.
(4) Reduction of sideways push with single-wheel bump or rebound at the rear end, and consequently, of rear-end steering effects.
(5) Better utilization of space at the rear of the car.

These advantages are not conferred automatically by the adoption of independent rear suspension: they have to be striven for, and it is most important that the designer should keep his aims clearly in mind. Some of the advantages may be obtained also with rear axles of the De Dion type.

The orthodox rear-axle layout, with semi-elliptic springs and Hotchkiss drive, is certainly less expensive at present and it is difficult to see how the price differential could be eliminated completely by development and simplification of independent-suspension systems. Of advantage to the orthodox rear-axle layout also is the accumulated knowledge of its problems and their solutions or palliatives.

Fig. 1 shows the independent rear suspension of the 2½-litre Lagonda which, like the Lancia Aurelia, has a pivot axis intermediate between fore-and-aft and traverse. Such a skew axis has an effective pivot or "swing-axle" centre where the axis crosses the plane of the rear wheels, so that variation is possible both of the "swing-axle" effect and of the height of the pivot centre. Lack of coincidence of the pivot point and the centre of the inboard universal joint implies "pumping" of the splined extension joint of the side shafts under bump and rebound, with additional friction in the suspension, particularly when torque is being transmitted.

In the swing axle rear suspension of the 3-litre Mercedes, normal resilience is provided by coil springs bearing on top of the half-axles, but each wheel is coupled also, by a long arm visible in Fig. 2, to a fore-and-aft torsion bar. The forward ends of the torsion bars are free under light-load conditions, the upstanding levers being clear of abutments. *Under full-load conditions, however, the abutments are brought into contact with the levers by the operation of an electric motor and intermediate gearing and screws. The swing axle is pivoted within about the first quarter of the distance between vehicle centre-line and wheel and, like a skew axis that is parallel to the ground, has no steering effect on bump and rebound.



A Drive unit
B Inboard brake drum
C Inboard universal joint
D Torsion rod
E Rubber spherical bearing
F Shock absorber
G Wheel-carrying arms
H Outboard universal joint
J Wheel attachment flange
K Operating arm for torsion rod
L Pivot axis
M Frame cross member
N Rubber spherical bearing

Fig. 1. Independent rear suspension of 2½-litre Lagonda

Unsprung weight

Both independent rear suspension and the De Dion rear axle permit unsprung weight to be reduced, principally because the driving unit is mounted on the frame but also because the coil springs or torsion bars usually provided are lighter, for a given duty and stress, than leaf springs. Moreover, less of their weight is unsprung. For a car weighing about 30 cwt unladen, the unsprung weight of the rear axle and appropriate parts of the semi-elliptic leaf springs and propeller-shaft was 195 lb per wheel. This figure compares with unsprung weights of 95 lb per wheel for I.R.S. and 111 lb per wheel for a De Dion axle incorporating torsion bar springing. In both examples the use of inboard brakes accounts for a saving of 35 lb per wheel. Reduction of unsprung weight results in a superior ride, and better road-holding on corners and during braking, because improved contact is obtained between road and wheels.

The figures tend to exaggerate the gain, however. The 195 lb of the more orthodox suspension applies to parallel lifting of the whole axle: the effective unsprung weight for single-wheel bump may perhaps be 10-20 per cent lower, k^2 for the axle being considerably less than the square of the half-track. Furthermore, the effective unsprung weight for an I.R.S. with swing-axle effect, allowing for angular acceleration, may in the worst case be roughly 8 per cent more than the actual unsprung weight cited. Approximately, this addition is inversely proportional to the square of the effective swing-axle radius.

Apart from reducing unsprung weight, inboard brakes are easier than outboard brakes to keep cool and are therefore less liable to "fade". With them, mechanical operation for the hand brake is simpler to arrange and is unaffected by relative movement between wheels and vehicle. Loads on the suspension linkage during braking are much reduced. If brake hop is to be avoided, limitations are imposed on the form of linkage that can be used with outboard brakes. A reasonable diameter of inboard brake drum can be accommodated, despite the need to maintain adequate ground clearance. Incidentally, the drive casing of a

*Précis of a paper presented by D. Bastow, B.Sc.(Eng), M.I.Mech.E., to the Institution of Mechanical Engineers in November, 1951.

car having I.R.S. or a De Dion axle can drag in soft, rutted ground without great loss of adhesion.

Rear-axle misbehaviour and propeller-shaft torque effects

Motions to which the normal rear axle is subject include parallel hop, with associated fore-and-aft movements and torque wind-up and from which are derived brake and power hops; and tramp, with associated yaw and sideways shake. As one wheel in an I.R.S. can affect the other only through the sprung mass, it seems likely that tramp and yaw can be eliminated and sideways shake reduced, but vibration problems have to be considered carefully. Brake and power hop are eliminated.

For a car with a rear axle, the propeller-shaft torque causes a difference in the weight carried by the two rear wheels of the order of 1 per cent at a steady speed of 50 m.p.h. and possibly 5-13 per cent with full torque in bottom gear. Although the maximum loss in adhesion of any one wheel is only about 3-6 per cent, therefore, the tendency to rear-wheel slipping, either sideways or rotationally, seems to be less for a car fitted with I.R.S. than for one with the normal rear axle. Possibly the improved "feel" is due to reduced unsprung weight and corresponding reduction of variations in road reactions, rather than to the elimination of the propeller-shaft torque effect itself.

With the normal rear axle the weight transference resulting from propeller-shaft torque, equal and opposite at the two ends of the car, itself produces an oversteering tendency on a right-hand corner and an understeering tendency on a left-hand corner. This effect can be neutralized by the use of a torque arm offset from the vehicle centre-line, but at the cost of unpleasantness during heavy braking.

Sideways movement of car due to one-wheel bump

The roll centre of the normal rear axle being above the ground, the tyre-points contacting the road are displaced sideways, Fig. 3, when one wheel of the rear axle is raised by passing over an irregularity of the road surface. As the axle assembly has sideways inertia, and as there is flexibility of attachment of the axle to the frame, the height of the effective pivot centre is intermediate between that of the geometric roll-centre and the axle's centre of gravity. The sideways displacement of the tyre is associated with a road reaction acting against the rear-end inertia and moving the car sideways in the direction reducing the tyre drift angle. The rear steering effect can be marked.

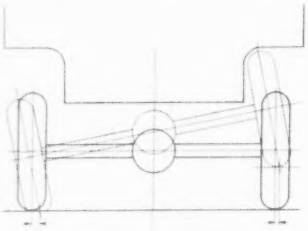


Fig. 3. Sideways movement of contact "points" with single wheel bump

With an I.R.S. having a roll-centre above the ground, however, only one tyre cannot point is displaced sideways and the associated side reaction is resisted by the other tyre as well as by inertia. If there were no inertia the rear-end movement of the car would be only half the contact-point displacement, so that the two tyres would have equal drift angles, but inertia reduces the rear-end movement to less than half. Neglecting the difference in inertia effects, Fig. 4 compares the half-displacement of one tyre in I.R.S. with the mean displacement of the two tyres on a rear axle (but the values are doubled to denote respectively the whole of the single displacement and the sum of the two displacements). The rear axle selected typifies present-day practice, and the independent suspensions have similar heights of pivot centre. Sideways push is reduced by lowering the roll-centre, and is influenced also, normally to a lesser extent, by any steering tendency of the rear wheels associated with wheel movements.

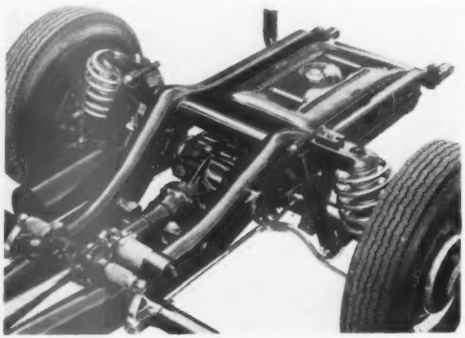


Fig. 2. Rear suspension of 3-litre Mercedes

It is undesirable to counteract the sideways displacement by causing the wheel to steer outwards as it moves toward bump, because: (a) although the drift angle is decreased on movement towards bump, it is increased on reverse movement; (b) it gives an oversteering tendency; and (c) variation in standing height with load must produce toe-in or toe-out in some conditions, and consequent wear of tyres.

Better space utilization at the rear

Frames have to be "kicked-up" to clear the rear axle and propeller-shaft in the full-bump position. With I.R.S., however, a frame close enough to the centre-line can pass under the drive shafts, and wheel-bump causes no corresponding lift of drive casing and propeller-shaft relative to the frame and body. Secondly, the propeller-shaft being fixed, engine and pinion axes can be made to point downwards at equal angles, as in the Lagonda, retaining constant angular velocity at the pinion. When a propeller-shaft moves with wheel-bump, engine and pinion axes need to be parallel. Again, the gear-box and clutch may, without increasing unsprung weight, be positioned on the final drive unit under the rear seat, in order to remove the hump from the front floor. This system is employed in the Lancia Aurelia.

Figs. 5 and 6 show how seat and boot layout can be modified to secure the additional accommodation that I.R.S. permits, the seat height above ground and the rear overhang to accommodate the spare wheel being unaltered. Setting the rear seat back over the wheel axle may be justified by the absence of rhythmic disturbances of the rear wheels and reduced unsprung weight. Seat and rear floor are low, and no propeller-shaft tunnel is necessary. The rearward seat position is possible only if the desired width of seat can be accommodated without resort to excessive rear track. For a given wheel track, the swing axle allows rather less distance between wheel arches than would an axle, but when the instantaneous pivot centre moves to the plane of the opposite wheel, the distance between wheel arches is greater than with an axle.

Whatever the design, an I.R.S. allows a lower boot floor-level. As illustrated, the boot floor is 4 in lower and the additional volume of boot thus obtained more than compensates for the

loss in volume due to the 7 in rearward displacement of the seat. It is assumed that the floor of the boot must be flat.

Requirements for acceptable handling characteristics

Certain cars with swing axle or parallel-motion I.R.S. have shown pronounced oversteer and a tendency to unexpected "break-away" at the rear end. On British roads, with many corners, less understeer is desirable than in a country where long straights and fast corners predominate. Excessive understeer means hard work. At slight yaw angles, the centre of lateral pressure of a relatively streamlined shape, without a tail fin, is far forward and introduces an oversteering tendency on fast corners. Whatever the degree of understeering desirable, it ought to increase progressively with increase in sideways acceleration, and rather more rapidly at extreme accelerations to give warning when break-away is about to occur. The converse characteristics can be most unpleasant. Break-away is influenced by bump and rebound buffers, and by the effect of traction which, reducing the cornering power of tyres, can be used to produce rear end break-away.

The handling characteristics of a car depend, under steady conditions, upon weight distribution; front and rear roll stiffnesses; heights of roll axes at the planes of the axles; extent to which the wheels roll with the car, and their normal attitudes or camber angles; the geometrical roll-steer characteristics; and tyre pressures. Handling characteristics for steady conditions can be calculated, given the tyre characteristics and appropriate information about the car. Fig. 7 depicts the calculated steering characteristics of a car weighing 33·7 cwt as tested and having front and rear weight distribution of 49-51 per cent, normal rear axle with roll centre 10½ in above ground, independent front suspension and front anti-roll rod. An understeer angle of 2½ deg at a sideways acceleration of 0·5*g* is reasonable, as is the form of the curve. Fully laden, the car would

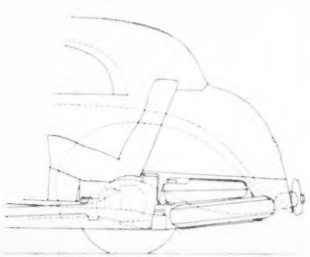
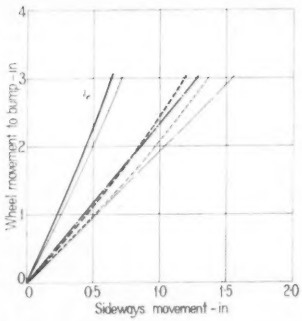


Fig. 5. Rear layout—car with I.R.S.



——— I.R.S.—instantaneous centre in plane of other wheel and 13·5 in high
— — I.R.S.—swing axle
— — Rear axle—sum of sideways movements of two wheels
Thick line—fully laden
Thin line—driver and one passenger

Fig. 4. Sideways movement of ground contact "point"—I.R.S. and rear axle

exhibit neutral steering throughout the range of sideways acceleration, but understeering can be restored by extra inflation of the rear tyres.

Olley has indicated ways by which an I.R.S. can be designed with the height of roll-centre and the position of the instantaneous centre of the suspension linkage independently variable. In cornering, roll-centre affects the weight transference, and the instantaneous centre of the linkage determines the extent to which the wheels roll. Tyre deflection and the "spragging" effect of the swing axle linkage (tending to lift the rear of the car because there is greater sideways force on the more heavily loaded outer tyre) being neglected, a linkage instantaneous centre on the car centre-line leaves the wheel angles unchanged in relation to the ground as the car rolls. An instantaneous centre at infinity, however, implies that the wheels will retain the same angle relative to the car, rolling with it. Wheels that roll with the car develop camber thrust outwards and so tend to oversteer. The "spragging" effect, characteristic of swing axles, also results in outward camber thrust.

A car with a swing-axle suspension arrangement having high roll centre and without anti-roll rod, but otherwise as in the example treated in Fig. 7, has undesirable steering characteristics with pronounced oversteer at a sideways acceleration of 0·5*g*. Calculated curves, both for a similar layout having an initial negative camber of 2 deg on each rear wheel, and also for a parallel-action independent suspension with rear roll-centre 6 in above ground, have this same undesirable feature, oversteering being about 1 deg at 0·5*g*. In Fig. 8, however, in which are shown steering characteristics of a similar car fitted with front anti-roll rod and having rear roll-centre 9 in above ground (giving approximately the same roll angle as a rear axle with roll-centre of 10½ in above ground), at least two of the curves for various positions of instantaneous centre are acceptable. Instantaneous centres are at distances, measured from a wheel plane, of ⅜ track (A); ½ track (B); ¾ track (C); in plane of other wheel (D); twice track (E), and infinity (F). Evidently satisfactory handling characteristics can be obtained without difficulty, but understeering in the curve for parallel motion drops away at sideways accelerations above 0·25*g*. For given roll-centre height, less swing-axle effect (angularity) implies less understeering.

Fig. 9 shows, for the car assumed, and with front anti-roll rod, the locus lines of linkage instantaneous centres for given understeer angles at 0·5*g*. A linkage instantaneous centre nearer to the wheel allows a higher roll-centre. Although the position of the locus lines must vary with the design of the car, the form of Fig. 9 holds in general. The extra sideways camber thrust for wheels which lean over further with the car must be compensated by less weight transference, if the drift angle of the rear tyre is to be maintained.

Since additional load tends to give oversteer, a reasonable amount of swing-axle effect should aid in preserving acceptable steering characteristics throughout the load range, because it tends to give negative camber which reduces oversteer as load is increased. The swing-axle radius needed to produce given camber change must vary with the stiffness of the rear springs, which determine the change in height for a given change in load.

The foregoing calculations were based upon pre-war tyre characteristics; present-day tyre constructions and wider rims have altered tyre characteristics but their general form is similar.

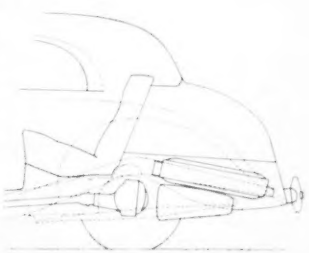


Fig. 6. Rear layout—car with orthodox rear axle

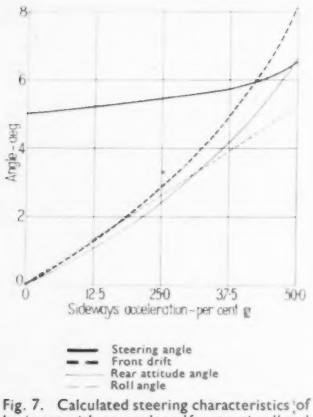


Fig. 7. Calculated steering characteristics of basic car with rear axle and front anti-roll rod

Concluding observations

With I.R.S. there is, on the one hand, reduced tendency to roll because the effective spring track is equal to the wheel track; on the other hand, increased tendency to roll results from the absence of the stiffening effect of half-elliptic leaf springs in twisting and sideways bending, claimed to be a 50 per cent increase. For a car with a rear track of 57 in and 44 in spring centres on the axle, however, the net effect is a reduction of roll which, as calculated for I.R.S. and anti-roll rod, varies in straight line fashion between $6\frac{1}{2}$ deg and $4\frac{1}{2}$ deg at $0{\cdot}5g$, with rear roll-centre between ground level and 15 in above ground. A rear-axle layout of

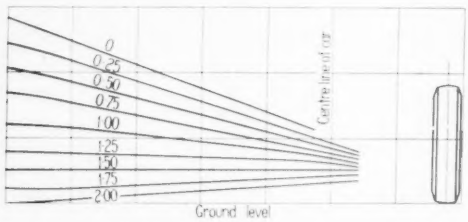


Fig. 9. Locus lines of position of instantaneous centres of linkage to give equal understeer angles—basic car with I.R.S.—at $0{\cdot}5g$ sideways acceleration

equal roll-centre height rolls approximately $\frac{1}{4}$ deg more at $0{\cdot}5g$ sideways acceleration. A car with I.R.S. could have its rear roll-centre $1\frac{1}{2}$-$1\frac{3}{4}$ in nearer the ground for the same roll angle.

Reduction of unsprung weight, and of the tendency to hops, tramps, and shakes, implies greater freedom in the choice of damping characteristics for a comfortable ride. While the ride on bad roads can be excellent, however, little bumps are sometimes brought into prominence with I.R.S., possibly because their effects are absorbed less by the tyres when unsprung weights are low. This stresses the importance of reducing friction to a minimum in the suspension linkage.

The desirability of allowing wheel movement between bump and rebound positions that shall be in keeping with the low spring-rates now sought, especially at the rear in view of the load variation, does not always seem to be appreciated. The noise of the final drive gears, mounted on the frame, can be made inaudible if they are accurately cut, rigidly supported, and flexibly mounted. There is no reason why I.R.S. should be less reliable than an orthodox rear-axle suspension. Rubber bushes have the advantages of requiring no lubrication and of reducing static friction in the linkage and the transmission of road noise to the body. They contribute spring rate without adding to load-carrying capacity, however, and care must be taken to avoid excessive stresses in the suspension springs themselves.

For the universally jointed side-shafts, the simple, reliable, and inexpensive Hooke's joint is desirable. It is eminently suited to the parallel-motion type of suspension, with which it gives constant angular velocity at the wheel. It is suitable for the De Dion axle, and has also proved so for suspensions with swing-axle effect provided the differential axis is in line with the wheel axes in plan view. A joint need be used only at the inner end with the swing-axle type proper. Any departure from alignment in plan view implies much greater variation in angular velocity with up-and-down wheel movement. It may amount, for example, to 1 deg for a 3 in rebound if the differential axis is 4 in out of line with the wheel axes. In these circumstances the Hookes' joint is satisfactory only for one condition of load, but a constant-velocity joint having the qualities of cheapness, reliability, reasonable size, and of being self-sustaining and in balance both when in line and when angularly displaced, does not seem to exist.

Frame members as wide apart as is now customary must have kick-up to clear the universally jointed drive shafts; only frame members located

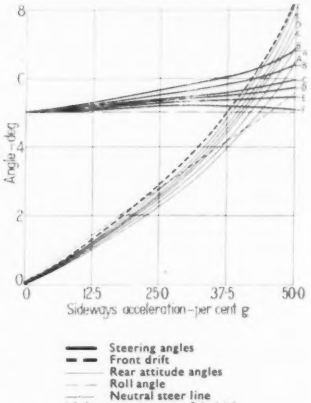


Fig. 8. Steering curves—basic car with front anti-roll rod

close together at the rear can pass beneath the drive shafts without loss of ground clearance to give the full advantage of I.R.S. With a close location, the attachment of the inner ends of the linkage can be more easily arranged. Frame members close together tend to be weak in torsion, however, and the gap between side members and body sills implies further loss of strength and difficulty in mounting the body. A new approach to frame design seems to be required. Unit construction of chassis and body may provide a solution but, owing to the tooling costs and development difficulties, this type of construction is usually reserved for cars produced in large quantities, and these are less likely to be the medium for the early exploitation of I.R.S. than are the more expensive cars catering for drop-head or soft-top bodies, and requiring a high degree of torsional stiffness in the frame. It is difficult to see what the final solution may be; ideas must prove themselves in production.

Furnace Control

DIFFICULTIES arise in maintaining accurate temperature control of electrically heated furnaces and ovens as a consequence of the wide variations, occurring at the present time, of both voltage and frequency of the supply mains. Sunvic Controls, Ltd., Essex Street, London, W.C.2 have now in production the RT.2 Resistance Thermometer Controller specially designed to overcome this problem. Arranged for mounting in standard P.O. racks, a neon lamp indicates the action of the load contacts and a bridge balancing potentiometer and sensitivity control are available on the front case. Two industrial type vacuum valves are used and no electrolytic capacitors are employed. (1992)

TYRE TESTING

Some Aspects in Relation to Vehicle Behaviour

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and P. W. B. Jones, B.Sc.(Eng), G.I.Mech.E.

This paper, presented to the Midland Section of the Institution of the Rubber Industry, opens with a brief statement of the essential requirements of a tyre. Road tests are necessary to ascertain whether pneumatic tyres meet these requirements and to make numerical estimates of the various properties involved. Frequently the results are influenced by the the vehicle and these influences have to be borne in mind in interpreting the findings.

No attempt is made to produce a general survey of tyre testing but instead selected probelms which have arisen are discussed in the light of the above considerations. Examples relate to cornering, skidding, comfort, effective rolling radius, tread wear and the estimation of wheel load.

IN addition to supporting load and transmitting tractive or braking effort, a tyre must also have the following features :

(a) An ability to provide a cornering force when operating at a drift or slip angle, that is, to set up a thrust perpendicular to the plane of the wheel when the latter moves in a direction making a small angle with the plane of the wheel (Fig. 1).

(b) A tendency for the plane of the wheel to revert to the general direction of motion. This manifests itself as a self-aligning torque when the tyre is running at a slip angle (Fig. 1).

(c) Ability to act as a "low pass filter", that is, to attenuate the high-frequency vibrations caused by road irregulatiies. The ability to absorb road irregularities demands a flexible road-contacting surface, or tread, which can deal with irregular departures of essentially a three-dimensional character from a true plane.

(d) A design of road contacting surface, or tread pattern, capable of causing the break-down and also removal by wiping action, of films of water or mud from the road surface.

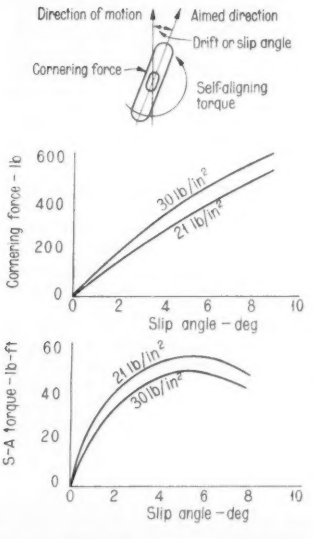


Fig. 1. Cornering force and self-aligning torque, 6.00×16, 10 cwt

(e) A tread material possessing a high coefficient of friction when in contact with existing road surface materials cleaned by the actions mentioned in paragraph (d). An important secondary feature is that the coefficient at very low rates of slip should exceed that in the truly static condition.

(f) A tread material sufficiently flexible to accommodate minor irregularities of road surfacing materials without introducing excessive road contacting pressures (resulting in crushing of the road material or other undesirable effects). These requirements (d), (e) and (f) are closely related for (d) sets a minimum value to the rigidity. If the tread material is a natural or synthetic rubber, however, requirements (e) and (f) are met automatically by the nature of the material, leaving (d) as the important one requiring consideration.

(g) A tyre structure being as free as possible from ill effects due to fatigue, self-generation of heat, permanent set, etc.

(h) A tread having good wear-resistant properties.

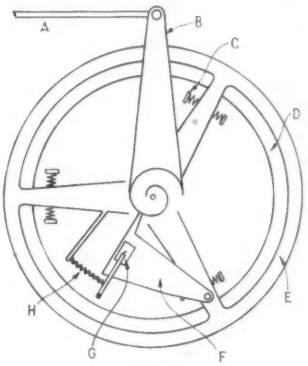
(i) Low weight, accurate concentricity and inherent balance.

(j) Readily adjustable spring factor and cornering characteristics, within limits, so as to match vehicle characteristics.

(k) It should not produce a characteristic vibration or noise when running on the road nor should it enhance any possible resonances in the vehicle structure.

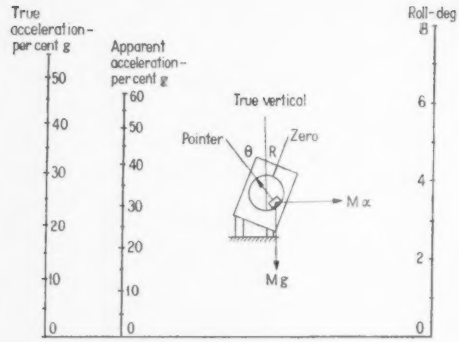
The first three of these items, namely cornering force, self-aligning torque, and absorption of road irregularities are considered to be of prime importance.

The pneumatic tyre meets these requirements in all major respects. In a practical design, however, some compromise between these several features is inevitable. In deciding the balance between them numerical assessments, by laboratory and road tests, become necessary. It can be seen that the field to be covered is very wide and a comprehensive paper is almost impossible. No attempt is made therefore to produce a general guide to tyre testing but selected problems are discussed in relation to the foregoing requirements.



A Anchorage
B Arm carrying cam
C Springs measuring force
D Backplate fitted to steering wheel
E Steering wheel
F Arm carrying recording film
G Recording stylus
H Push rod transmitting spring deflection

Fig. 2. Steering wheel to record characteristic force-movement curve



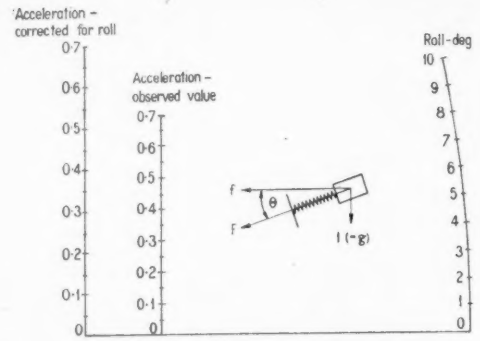
$\text{Tan } \Theta = \frac{\alpha}{g} = Y$ = true acceleration

Observed angle $= \Theta + R$

Apparent acceleration $= T = \tan(\Theta + R)$

$\therefore \tan^{-1} T = \tan^{-1} Y + R$

Fig. 3a. Nomogram for correction of mass on pendulum accelerometer



$= \frac{F - \sin \Theta}{\cos \Theta}$

This is equivalent to a zero shift of g sin Θ and a change of scale dependent on cos Θ

f = true acceleration – g

F = recorded acceleration – g

Θ = tilt due to roll – deg

Fig. 3b. Nomogram for correction of mass on spring accelerometer

Cornering characteristics

The typical cornering force curves are well known and were given in the first instance in Refs 1, 2, and 3, and some of the consequences on the road are further discussed in Refs 4, 5, 6, 7 and 8. It is not proposed to discuss the direct measurement of cornering force on rigs but rather the assessment of the behaviour of a vehicle when cornering with tyres of known characteristics. One method of tackling the problem is the checkerboard test described by Stonex, Ref 9, but the method is, unfortunately, too elaborate for ordinary usage. A number of alternatives have been explored with varying degrees of success. In this work, the primary aim is to achieve the measurement of the vehicle and driver's reactions to an oversteered condition. Oversteer is the unstable condition where departures from a straight or uniformly curved course cause forces at the tyre-ground contact which aggravate the departures from the intended course.

It is perhaps worth placing on record that Hardman, Ref 8, has pointed out that the usual definition of oversteer, that is, the rear slip angle is greater than the front, applies only to a straight path. A more general definition can be stated in terms of radius of curvature or curvature. In a neutral steered vehicle the curvature of path of the centre of gravity is unaltered by changes in external side force. If the concavity of the curvature viewed from the side of application of the increase of external force is increased by the change, then the vehicle is oversteered. The easiest way of producing oversteer intentionally is a reduction in rear tyre pressures, this results in a lower cornering power at the rear than at the front.

There are two main ways of investigating this problem, one is to measure vehicle course and the other to measure steering wheel motion. One well-known method of determining the vehicle attitude angle, that is, the angle the vehicle centre line makes with a tangent to its curved path, is the skid pad, Ref 5, but work is restricted to steady turns of constant radius. Tests of this kind do not lead to the desired information. An alternative to this technique has been described by Hardman, Ref 9. In this method the radius of the path of each wheel is measured and the slip angles and hence the degree of over or understeer derived from the known geometry of the vehicle. For practical reasons this method is also restricted to the study of steady motion on a circular course of relatively small radius.

The important effects of oversteer arise, however, when the vehicle straightens-out on leaving the curve. These effects arise from time lags and they are analogous to time lags in any servo-mechanisms. Measurements on a directional gyro show that the departures of attitude angle from a simple transition curve are of the order of 1 deg or less even when the oversteer was easily detected subjectively. The time lags could not be estimated from the results. Thus the method will require more sensitive instruments to be of practical use.

The reaction of a driver to a condition of oversteer manifests itself in his actuation of the steering wheel irrespective of how he judges the vehicle to be oversteered. Separate records of movement against time, or

Fig. 4. Car rolling on a bend

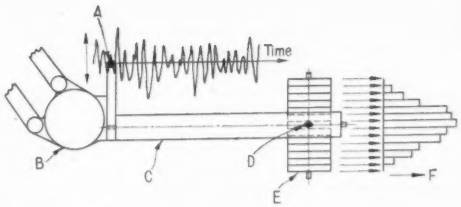
force against time, do not lead to a satisfactory assessment of this oversteer correction. Records of force against time are, however, very useful for assessing the degree of self-aligning torque and other torques affecting the applied steering effort. It has become clear that to study oversteer the characteristic steering force-steering movement curve for the vehicle in various manoeuvres is needed.

No doubt this diagram will reveal many other features but with a good steering gear, free from excessive backlash and compliance, it should reveal a different characteristic curve for oversteer from that for an understeered vehicle. An experimental device somewhat as in Fig 2 has been tried and the results to date appear promising. This work is still in an experimental stage and the device is to be modified to alter the relative scale values of the diagram to ease interpretation.

Skidding characteristics

The methods used for straight ahead skidding and braking tests have already been described in Ref 11. It is sometimes desired to measure the sideways skidding characteristics of a tyre in order to check the relationship with the straight ahead coefficient. To eliminate the influence or control the driver may have over the result, the best procedure was found to be as follows :

The recording accelerometer is placed in the vehicle as for braking tests but suitably oriented to measure sideway acceleration. The car is driven on to a suitably wetted, smooth test track at 30 m.p.h., the gear disengaged, and the steering wheel turned "hard over" to full lock as rapidly as possible. The vehicle develops a broadside skid. The actual course on the road surface is settled by the inward accelerations imparted by the frictional grip between tyres and and so the accelerometer records the inward acceleration. Because the vehicle "rolls", a correction must be made to the acceleration diagram. The nature of the corrections, and nomograms to facilitate their calculation, are given in Figs 3a and 3b.

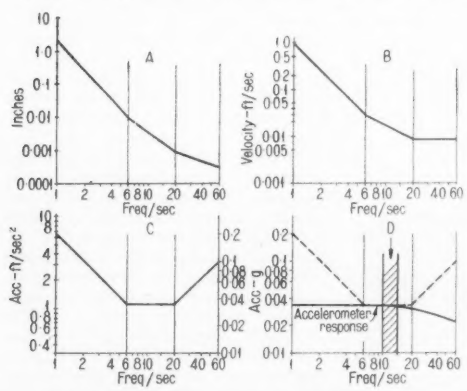


A Tracing point
B Standard drafting head
C Roller maintained parallel to time base
D Roller rests on disc
E Bank of recording discs
F Amount turned

Fig. 6. Principle of distribution analyzer

There are several ways of measuring the roll angle. The roll gyro and the horizon camera have already been described, Ref 11, but a simpler and, perhaps, more direct method is to photograph the vehicle from a point outside the vehicle chosen so that the view is approximately along the longtitudinal axis, as in Fig 4. A typical roll is 10 deg per g. In straight comparisons of tyres on a given vehicle there is of course, little need for this roll correction. In passing it is of interest to record that the pitch in straight ahead braking tests is considerably less—0·5 deg to 1·0 deg per g is typical—and there is little need for this correction in braking tests.

The sideway and forward coefficients are in broad correspondence. The former, however, is around 10 per cent lower than the latter on a smooth test track when wet. Such factors as influence cornering power would, perhaps, be expected to cause an improvement in the sideway value. Tests suggest a tendency in that direction but the effect is small and it is difficult to establish any real difference. This is due to the effects, if present, being of the same order as the day-to-day test variability.



A Amplitude of Displacement
B Amplitude of Velocity
C Amplitude of Acceleration
D Unsprung mass frequencies

Fig. 5. Janeway's recommended limits (A, B, C). Justification of use of accelerometer (D)

Comfort

As regards vibration, a car is a very complex system. There are however two main frequency ranges which are of interest from a comfort aspect, Ref 12. They are (1) the body and chassis mass on the suspension springs and (2) the axle mass on the tyres. The frequency ranges are typically 60-80 per min and 10-15 per sec respectively.

From his survey of the literature on comfort, Janeway, Ref 13, recommends that the upper limits of amplitude of vibration for satisfactory comfort of a vehicle are :

Frequency Range c/s	Formulae for limits	Criterion
1–6	$af^3=2$	Jerk
6–20	$af^2=\frac{1}{3}$	Acceleration
20–60	$af=\frac{1}{60}$	Velocity

where a is amplitude (in) and f is frequency (c/s).

Fig 5a shows a plot of these limits in terms of amplitude against frequency. Figs 5b and 5c show the same curve replotted as velocity and acceleration respectively. It follows that acceleration is a useful index of comfort providing :

(1) The frequency range of the oscillations being investigated is between 6 and 20 c/s.
(2) The accelerometer employed records the acceleration without serious distortion.

According to usual practice, Ref 14, this is so if (a) the natural frequency of the instrument is about three times the frequency of the oscillations and (b) the damping is around 0·7 critical.

It seems reasonable from Fig 5d to use an accelerometer to assess the relative comfort of different tyre fitments, provided the test road does not set up large chassis movements. The accelerometer gives an acceleration-time curve which needs summarizing before simple comparison can be made.

A complete summary of the amplitude changes is obtainable from the distribution analyzer, already described in Ref 11 and sketched in Fig 6. A disc's rotation records the proportion of the total time during which the amplitude lies between the two values corresponding to the edges of the disc. These values then give

the distribution of the recorded accelerations. To further condense the information it is desirable to obtain the standard deviation, which is a statistical measure of spread, in acceleration units. The mean acceleration is always zero as the vehicle, on the average, remains on the ground. This can be done by plotting the cumulative totals on probability paper and drawing the best straight line, Ref 11. This corresponds to fitting a normal or Gaussian curve to the data and permits the standard deviation to be determined from the slope of the line.

Another way of summarizing the data is to calculate the root mean square (R.M.S.) value by summing the products of the disc movements and the square of the corresponding acceleration values, dividing the sum by the total of the disc movements, and finally extracting the square root. It is seen that this is also the process of finding the standard deviation and these two measures are identical although the two ways of estimation described may not lead to identical values. A consideration of these relationships has lead to a quicker method of estimation.

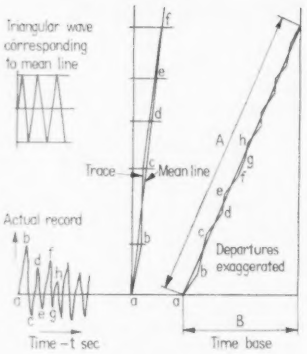
If the predominant frequency of the record is known, a quicker way of comparing vibrations is to estimate the average peak amplitude by the following method. A map-measurer is traversed along the acceleration trace and also along the true base, the recorded values being A and B respectively. The average peak value is then approximately

$$P = \frac{1}{4\ f.t.s}\sqrt{A^2 - B^2}$$

where

f = dominant frequency, axle mass on tyre as spring.
t = duration of record (sec)
s = scale factor of acceleration diagram relative to map measurer units in number of units per *g*.

This expression is exact if the wave form is triangular and of constant amplitude and frequency. Fig. 7 shows that departures from the triangular wave form do not seriously alter the estimate. A straightforward calculation shows that the peak value is $\sqrt{3}$ times the R.M.S. value in the case of the triangular waveform. The use of such a factor to obtain a figure comparable with the S.D. or R.M.S. values is merely equivalent to fitting triangular waves to the vibrations recorded by the accelerometer. In practice, this approximately derived relationship is adequate provided the predominant frequency is well defined and the greater speed of this new method is useful. (See also appendix.)



$\sqrt{A^2 - B^2} = 4$ fts (av. peak value)
f – predominant frequency
s – scale factor – in *g*

Fig. 7. Simple method of estimating average peak value

The foregoing suggests that the tyre alone controls the vibrations of the 10-15 per sec frequency which give rise to a feeling of roughness on, say, rippled road surfaces. Unfortunately this is not so; resonances or interaction between axles and, or, other spring mounted components such as the engine may either magnify or absorb the initial axle vibration.

Fig 8 compares two vehicles at each of two speeds. In one there is no interaction at either speed but the other is rendered less comfortable by interaction of some kind, which occurs at 25 m.p.h. when the rear axle hits the obstacle before the vibrations set up by the front wheel have died down. The vibrations continue to increase after the rear axle hits the obstruction.

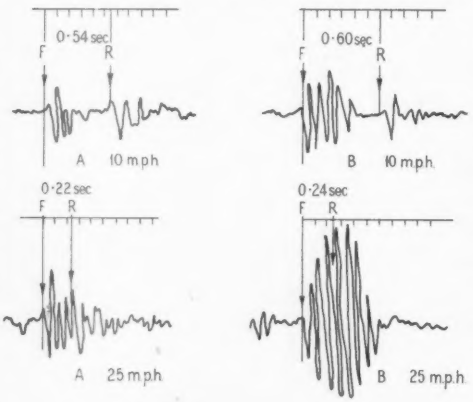
Another feature which often escapes attention is that the shock absorber, although connected to the chassis and not the ground, provides the damping which controls the high frequency axle vibrations as well as the lower frequency, 1 per sec, body oscillations. This can be seen from Fig 9 for, at the frequencies of 10-15 per second, the chassis mass is practically stationary because of the relatively lower natural frequency of its motion on its support. Hence the shock absorber motion has approximately the same amplitude as it would have if it were connected from axle to ground.

Because the inherent damping of the tyre is small, it follows that the small movement, high velocity characteristics of a shock absorber have a very important influence in tyre testing from the comfort aspect. Comparisons between tyres must therefore be made on the same vehicle over the same stretches of road. In general, provided no resonances occur in the vehicle, the comfort of a tyre on a given vehicle is in the same relative order as the cushioning capacities, the inverse of the spring rate.

Effective rolling radius

The number of revolutions a tyre executes in a given distance at its scheduled operating conditions requires to be known for the purposes of speedometer design. It is common to call the radius which may be derived from the revolutions-per-mile figure the rolling radius. It is not equal to the height of the axle from the ground nor is it the radius of any identifiable point of the tyre.

It is an effective radius, and can be estimated from tyre dimensions by empirical formulae which, in general, are of the form: effective rolling radius equals standing height plus a certain percentage of tyre deflection. The percentage depends somewhat on tyre construction and is around 20-30 per cent. These formulae are based on observed values of revolutions per mile under known running conditions. Revolutions per mile are easily determined in the normal driving case by running the vehicle over a suitable measured distance, preferably 1 mile straight, and counting revolutions by mechanical or electrical counters, Ref 11.



A Good vehicle
B Car with interaction between axle vibrations
F Front wheel hits plank
R Rear wheel hits plank

Fig. 8. Vertical acceleration when riding over a ¾ in plank

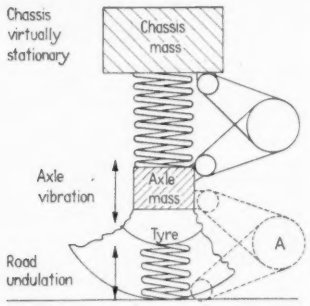


Fig. 9. Chassis is seismic at axle resonance frequency. Shock absorber moves as if connected at A

A high precision is possible. The effective rolling radius is altered slightly by torque transmission, the effective radius being greater for driving, to provide for the driving slip, and likewise less for braking.

In four-wheel-drive vehicles the effective rolling radius is of particular importance as it settles the distribution of transmitted torques between the front and rear axles which, of necessity, execute the same number of revolutions when coupled. Some interesting effects arise. For example, it has been found that a particular four-wheel-drive vehicle with heavily loaded, underinflated rear tyres, giving smaller effective radius, and lightly loaded front tyres, giving larger effective radius, resulted in heavy heel-and-toe wear in a braking direction on the rears and little wear on the fronts. This, together with the fact that one turn of the front wheels corresponds with a longer distance than one turn of the rears, suggested the apparently paradoxical situation that lightly loaded front wheels were pulling the heavily loaded rears around. Direct observation of the tyre motion on the ground proved this to be so and also showed the overall braking slip at the rears to considerably exceed the overall driving slip at the fronts.

The explanation of this phenomenon is contained in Fig 10, where it is seen that a driving wheel lays the tread down, on entering the contact area, in the way required for the normal deflected condition, that is, in compression. A braking wheel, however, requires movement within the contact area to change tension into compression. This provides the basis for an explanation of the greater torque transmission available from a driving wheel relative to a braking wheel. The diagrams can be shown to be similar for smooth wheels; the toothed wheels are used to make the facts clear. Furthermore, the characteristic coefficient of friction-slip curve, Refs 15 and 16, with its rapid increase to a maximum from zero to around 10 per cent slip, with a definite fall to a somewhat lower value at higher slips, serves to ensure that the braking tyre continues to slip and fails to cause the driving tyre to exceed the slip corresponding to the maximum grip.

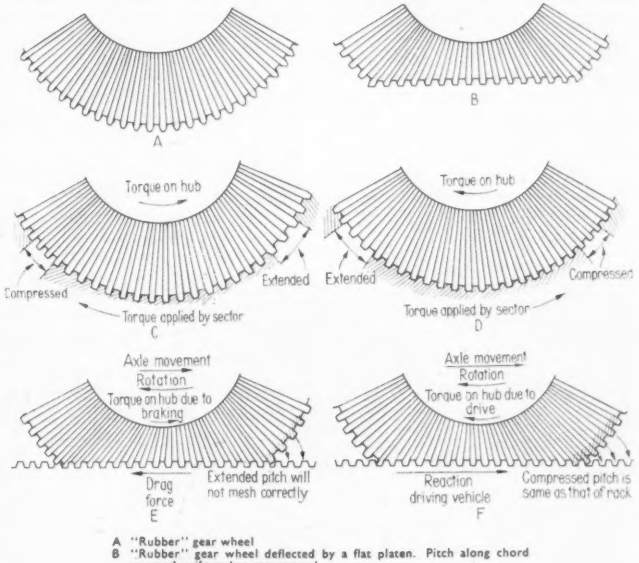
Tyre wear

There are many manifestations of patchy or irregular wear on tyres, particularly those on front wheels of vehicles. The two main types are probably scalloped wear and rib-punching, that is, excessive wear of one casing rib of the pattern. Scalloped wear does not arise from wheel flap as might be expected. For example, if a tyre with a 5 in or 6 in pitch scalloped wear traverses the road at 30 m.p.h. the frequency of flap would need to be 100 per second, which would be impossible. Again, if it depended on a simple mass-spring system it would be very speed-dependent and could be eliminated by avoiding the resonant speed.

The explanation of these phenomena rests in the fact that unequal lengths of tread material, due to differences in rib radius, are forced to travel the same distance along the ground or else slip. In fact, one rib progressively increases its longitudinal compression and the adjacent rib increases its longitudinal tension until the frictional grip between the ground and one or other of the ribs fails to be adequate. Slippage occurs at this point and then the cycle recommences.

If the points of slippage are randomly disposed around the periphery of the tyre, uniform wear results. If, however, due to a local depression, the slippage always occurs at a given point, and because the co-efficient of friction drops with the sliding, the tensions and compressions are completely released. Wear is accentuated at that point and, furthermore, the tensions and compressions always start rebuilding from that point. As the cycle is now phased to recommence at a fixed point, the succeeding slippage will also occur at a constant place. It is seen that once a flat spot is established others automatically develop around the periphery. The process is both self-generating and self-sustaining as any given flat will cause a further flat beyond it.

The original flat spot can arise from a wide variety of causes such as eccentric brake drums, accidental locking of wheels, discontinuities in tread rib longitudinal rigidity, or tyre repairs. If there is an overriding slip (sideways as in cornering or fore-and-aft as in driving wheels) there is no cyclic build-up of differential stressing, and irregular wear rarely arises. It follows that a freely rolling tyre is most prone to irregular wear, and also that the front wheel further



A "Rubber" gear wheel
B "Rubber" gear wheel deflected by a flat platen. Pitch along chord assumed uniform but compressed
C Clockwise torque applied to short sector of periphery
D Anti-clockwise torque applied to short sector of periphery
E "Rubber" gear rolling along rack with braking torque applied
F "Rubber" gear rolling along rack under the action of a driving force

Fig. 10. Explanation of effective rolling radius phenomena

from the steering box is most prone to irregular wear as this wheel is not called upon to give its fair share of cornering force because of slack in the links.

"Rib punching", where one rib wears faster than its neighbour, arises from similar causes but is accentuated by the presence of a high ground-contact pressure on shoulder ribs. Here again the findings on the four-wheel-drive vehicle apply, leading to a partial explanation as to why the crown rib wears least. It will be noted that the buffing marks on the tread confirm that at the points of high wear the rib moves in a longitudinal direction in accord with this theory. Fig 11 shows an example. From the buffing marks it can be shown that the place of heavy wear has occurred due to the rib moving in a braking direction on the ground. Elsewhere on the tyre the buffing marks are normal for a front wheel. The fine lines, which are normal to the direction of buffing, form an arrow-head pattern pointing upward when the tyre is viewed from the front of the vehicle. This latter pattern can also be shown to arise from the simple superposition of overall sideslip to the right and the left, on the fore-and-aft motions due to the different rib radii.

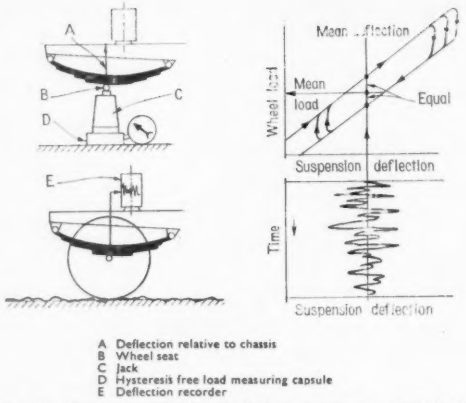
The same basic theory provides an explanation for the patchy wear of twins and also for the apparently more difficult phenomenon of hollow wear on wide, shallow, solid tyres. The foregoing indicates that only vehicles in good mechanical condition should be used for rate-of-wear tests, particularly when comparing tread compounds. Even when the vehicle is in good mechanical order and there is no sign of irregular wear, the vehicle design can influence the relative rates of wear at each wheel. For example the differences in the compliance of the steering linkages to each of the front wheels can seriously modify the slip angles.

Fig. 11. Tyre with irregular wear. Low spot and direction of rotation indicated by arrows

By theoretical analysis of the motion of a vehicle around a bend, having due regard to the steering gear compliance, the following example emerges :

Vehicle—R.H. drive saloon. 5·50—16 tyres. Compliance to wheel nearest steering box one half that to other wheel. Overall compliance from wheel to wheel is taken as 20 lb-ft per deg. Progressing at a uniform speed of 45·6 ft/sec around a right-hand bend of 225 ft radius, the rear wheel slip angles are each within a minute or so of 3 deg but the right-hand front wheel, inside the bend, has a slip angle of 4 deg 9 min while the left-hand front wheel, outside the bend, has a slip angle of 2 deg 57 min. The right-hand front wheel is developing a cornering force of 300 lb and the left-hand of 225 lb. Clearly the tyres will wear at different rates. To traverse this curve the steering wheel is turned through an angle which corresponds to an expected steered angle at the wheels of 4 deg 26 min but due to the compliances the right-hand wheel turns 3 deg 7 min and the left-hand only 1 deg 57 min. At these small angles the question of departure from Ackermann hardly arises. This result is typical of the findings of the analysis which is new and is helping the practical findings of wear tests to be understood.

Before leaving the important question of wear tests, attention is drawn to a recent paper by Buist, Newton, Thornley, Ref 17, and to record also that use is made of the statistical plan known as the incomplete randomised block. There are, however, two slight differences in analysis which should be put on record. It is preferable to analyze logarithm of wear rate rather than mm per mile or miles per mm.



A Deflection relative to chassis
B Wheel seat
C Jack
D Hysteresis free load measuring capsule
E Deflection recorder

Fig. 12. Principle of reliable method of estimating wheel load

This avoids any unnecessary discussion of the relative merits of the two ratios as their logs are numerically equal, though of opposite sign. More important, however, is that the analysis is more discriminating as the log of wear is more normally distributed than either of the two ratios. Further, impossible values such as negative rates of wear cannot arise in the analysis. The use of logarithms has proved invaluable in tyre life problems of other kinds.

The other pertinent point is that the analysis of variance should be set out with the principle of split-plot designs in mind, and arranged so that every term in the table corresponds to some physical effect. The result is that variabilities between tyres are not mixed with the smaller variabilities within a tyre. A paper has been read on this subject by Martin, Ref 18. Although statistical analysis is not directly mentioned elsewhere in this paper it is regularly used to help decide the significance of test results in connection with all matters investigated.

Measurement of wheel load

In any tyre test the wheel loadings are required to be known. It is common experience that weighbridge and, or, loadometer figures for individual wheels often fail to total up to the correct overall weight. The following procedure does lead to the correct answers, although it is too elaborate for any but special cases. It is included because it shows the nature of the difficulty and it demonstrates that it is not always safe to assume that if the load measured on an individual wheel repeats satisfactorily the value is the true operating load.

These effects are most marked in

heavy vehicles, especially military vehicles such as tanks. The procedure is as follows. The wheel is supported by a jack resting on a good quality, load-measuring capsule and the wheel is raised and lowered, over an adequate range, while deflections relative to the chassis are recorded. The results are found to be as in Fig 12, the width of the loop depending on spring friction, etc. The vehicle is then driven around over a typical road and a curve of axle movement relative to chassis recorded. The mean deflection read off the latter curve then enables the mean load to be found from the hysteresis loop.

The use of a single loadometer can give consistent but inaccurate results. Friction in the weighbridge can have a similar influence and so cause trouble. If the principle behind this method is borne in mind, apparent anomalies in results obtained by the normal weighing processes can usually be elucidated. It also applies to almost any measuring process leading to the general law that any measuring device with hysteresis can give consistent but biased results unless the measurements are performed in such a manner as to exhibit and assess the magnitude of the hysteresis.

Acknowledgments

Thanks are due to the Technical Director of the Dunlop Rubber Company for permission to present this paper. We must also acknowledge considerable assistance from all our colleagues, particularly J. H. Hardman who has made useful criticisms.

APPENDIX

General assessment of comfort

A possible extension of the technique of estimation of average peak value by the use of a map-measurer is the assessment of discomfort arising from chassis movement. Over the 1 to 6 c/s range, Janeway gives a f^3 as the index of comfort with a $f^3=2$ as the limit. This index can be written as $f(af^2)$.

If tests have to be carried out on roads with long and short waves present, it is suggested that the record can be interpreted by putting an average line through the record, any undulations on this average line to have a frequency of 6 c/s or less. Let the length of this line be C, then :

$$P_1=\frac{1}{4\,f.t.s.}\sqrt{A^2-C^2}$$

and $$K_1=\frac{1}{4\,t.s}\sqrt{C^2-B^2}$$

permit approximate comparisons of both unsprung and sprung oscillations but values of P_1 and K_1 cannot be directly compared as their relative importance is somewhat dependent on the observer preference. K corresponds to "Jerk" in Janeway's terms.

If, however, the view is taken that a scale of discomfort in which Janeway's recommended limits are unity everywhere, the following technique would be applicable in appropriate circumstances :

Find K in in/sec³ from A, B and C, t and s from the acceleration record, as already described, and also determine f, the predominant frequency in cycles per second, then :

If f is between 1 and 6 $J=\frac{K}{2}$

If f is between 6 and 20 $J=\frac{3K}{t}=3P$

If f is between 20 and 60 $J=\frac{60\,K}{f^2}$

J is then the comparative discomfort in a scale in which the recommended safe limit of vibration, as defined by Janeway, is unity whatever the frequency. The extension to the 20 to 60 range is probably not practicable because of the difficulties of finding a suitable accelerometer. The use of an instrument reading directly in velocity is probably preferable in this this region.

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HOT-SPOT MACHINING

ACCORDING to S. Tour in *Metal Progress*, June, 1951, the rate of metal removal in machining carbon-, chromium-, nickel- and stainless steels can be increased three or four fold by raising the temperature of the stock to between 500 and 1,500 deg F. The force exerted on the stock by the cutting tool is utilized to shear the stock, to deform the chip and to push it across the top of the tool. The energy absorbed is a function of the shear strength of the material which, for most steels, falls off rapidly at temperatures above 500 deg F. The tools are of the inserted carbide tip type and are uncooled. As carbides show only a moderate loss of hardness within the temperature range below 1,500 deg F, it is possible to extend the life of the tool, in spite of its reduced hardness and greatly increased cutting rate, by assigning it a reduced cutting load.

High frequency induction heating is selected for simplicity and ease of control, and is applied as fast as the tool moves. The temperature is maximum on the surface of the work and may be a few hundred degrees lower at the base of the cut. An exact temperature of the work is not essential. Although the tool receives no cooling, in order to retain the elevated temperature of the stock, coolant is used on the work after the cut is taken to minimize the expansion against the tailstock of the machine.

An additional advantage of the hot machining technique is in the finish of the work. Crevices appearing on the surface when a heavy cold cut is taken are absent. Feed lines are formed, but they travel around the periphery, while transverse tear lines do not arise because hot steel has no tendency to cold fracture. (*M.I.R.A. Abstract No.* 5514.)

PNEUMATIC TOOLS

Important Developments by Desoutter Brothers Limited for Assembly Functions

MANY interesting and important developments have been incorporated in the new assembly factory recently opened by Austin Motors Ltd. In general, the impetus for these developments came from the Austin planning engineers, but their successful application has been made possible only through the ready cooperation of several manufacturers of specialized equipment. Some of the developments in this new factory are so spectacular that others, of great importance but far less spectacular, may easily be overlooked. For example, it is doubtful whether the importance of the pneumatic tools developed and supplied by Desoutter Brothers Limited, The Hyde, Hendon, London, N.W.9, has been appreciated by other than a few of the engineers intimately concerned with the whole project.

Hitherto, almost without exception, automobile manufacturers have practically always used fairly heavy weight high-cycle electric nut runners for chassis and body assembly functions. On the new Austin assembly lines, however, pneumatic tools are used exclusively. This is, we believe, the first automobile assembly installation in the world on which pneumatic tools are used to the complete exclusion of electric hand tools. The primary reason for this development was that the assembly sequence laid down by the planning engineers called for the simultaneous tightening of several nuts at certain stations on the assembly line. Incidentally, certain other advantages accrue from the use of pneumatic tools.

Fig. 1. Desoutter SR.50 pneumatic motor. It is $1\frac{7}{8}$ in diameter, weighs only 2 lb 9 oz and develops nearly $\frac{3}{4}$ h.p.

Normal high-cycle electric tools with nut capacities of $\frac{1}{2}$ in and $\frac{5}{8}$ in are very bulky, have an average weight of 18/20 lb, and can be used for tightening only one nut at a time. Obviously, it is not practicable to combine such tools into a unit for tightening say four nuts simultaneously. Therefore, when the problem was placed before Desoutter Brothers Limited, that organization realized that the only practical solution was the use of pneumatic tools. It was also realized that for weight and space reasons, a new form of motor had to be developed. Finally, after considerable research, the SR.50 pneumatic motor was developed.

The SR.50 motor

This pneumatic motor, which is illustrated in Fig. 1, is only $1\frac{7}{8}$ in in diameter and $6\frac{1}{2}$ in long, and weighs only 2 lb 9 oz. Despite its small size it develops nearly $\frac{3}{4}$ horse-power with

Fig. 2. Using a twin motor unit for tightening two rear axle U bolts simultaneously

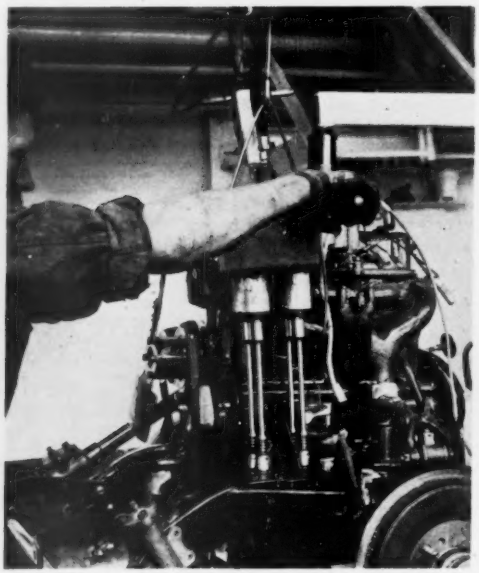


Fig. 3. A quadruple unit in use for tightening four nuts on the front mounting for the engine

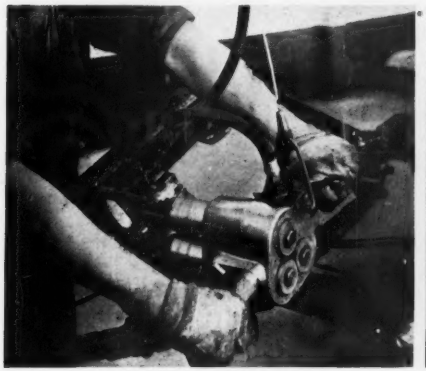


Fig. 4. Three SR.50 motors grouped for fastening the brake link bracket

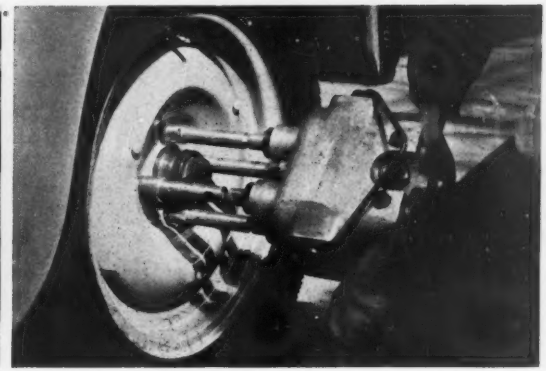


Fig. 5. A quadruple unit with SR.50 motors for tightening the wheel nuts

a torque of 500 inch pounds. Because of the small diameter, these motors can be grouped with spindle centres as close together as 2 in. Furthermore, owing to the low weight, an assembly of multiple spindle tools is light in weight, easy to handle, and the operator does not feel any torque reaction. This motor has been specifically designed for nut and stud running.

The rotor is mounted in ball bearings, but also rides on a cushion of air at each end. This arrangement eliminates friction and wear. It has five blades, which are automatically blown out when the motor is switched on, thus ensuring perfect starting. The case has a screwed nose and is therefore, suitable for end or clamp mounting in assembly fixtures. Two trains of epicyclic gears mounted on roller bearings provide a free spindle speed of 350 r.p.m. There is no clutch since the motor is designed to stall against a controlled air pressure.

That the motor stalls against a controlled air pressure is a most important feature of the design. It ensures complete uniformity of tightening torque. Before this development, most high-cycle electric tools used for running up and tightening $\frac{1}{2}$ in and $\frac{5}{8}$ in nuts on automobile assembly lines, were fitted with either spring-loaded or impact-type clutches. Despite the efficiency of such tools, the results obtained are in some degree dependent upon the operator, since they are effected by the amount of dwell allowed on each nut before the tool is switched off. With the SR.50 motor, the tightening torque is applied completely automatically, and it can be said with certainty that nuts can be tightened to a pre-set torque with unfailing regularity.

In an installation of the type employed by Austin Motors Ltd., it is essential that there is a compressed air system used solely for the supply to the pneumatic tools. Additionally, the main airline pressure must constantly be maintained at a pressure in excess of that required when all the motors are in use together. For this reason Austin Motors Ltd. have installed a special battery of three compressor units to supply air only for the pneumatic tooling equipment on all the assembly lines in the new factory. Air is supplied to all the main assembly points at approximately 150 lb per square inch pressure. The exact pressure required at the various points is obtained through individual pressure regulators at the take-off points for the various groups of motors. At each point the pressure is adjusted to give the correct degree of tightening torque for the job that is to be carried out. Provision is made for the lubrication of the motors by means of automatic air-line lubricators.

For an installation of this character, where a good supply of clean, dry air is assured, together with constant lubrication, pneumatic tools have other advantages than the facility with which they can be grouped to allow several nuts to be tightened simultaneously. For example, in comparison with electric tools, they should have a longer life, be practically free from breakdowns, and upkeep costs should be greatly reduced. It is, in fact, almost impossible seriously to damage a pneumatic tool through ill use. Incidentally, the SR.50 motors in use at the works of Austin Motors Ltd. are the only ones in existence. They are going into general production in the near future so that supplies should soon become available for other manufacturers.

Fig. 6. Tightening the rear bumper bracket with an SR.10 pneumatic spanner

Fig. 7. An SR.10 pneumatic spanner with long reach extension for fastening radiator support bracket

Fig. 8. Part of an Austin assembly line, showing multiple tools suspended from eight-armed windmills

Applications

Multi-spindle tools with SR.50 motors are used at five stations on the assembly line. They are arranged to stall against a controlled pressure which ensures that the specified degree of tightness is obtained on each nut. These stations are :—

Engine mounting brackets	4 nuts
Front assembly unit (shock absorbers)	4 nuts
Wheels	4 nuts
Rear brake link bracket ..	3 nuts
Rear spring shackle bolts..	2 nuts

Four of these multiple spindle tools are illustrated in Figs. 2, 3, 4 and 5. The tools shown in Fig. 2 is a specially-designed angle-drive twin motor unit for tightening two rear axle U bolts simultaneously. The motor performance is unaffected by the peculiar mounting, and the spring-loaded sockets make location of the unit upon the fixing bolts a very simple matter.

Fig. 3 shows a unit that has been designed for tightening four nuts on the front mounting for the engine. Each of the four spindles is fitted with a spring-loaded box spanner on a long-reach extension. This illustration also shows the method by which the special unit is poised over the work on a spring-loaded cable winch. The unit is attached to the cable by a suspension bracket, and in operation is balanced by its own weight. A triple unit for simultaneously tightening the nuts that fasten the brake link bracket to the chassis frame is illustrated in Fig. 4, and a quadruple one for tightening the wheel nuts in Fig. 5. In each of the multiple spindle units a single master trigger valve controls all the motors. In addition, the exhaust from the motors is taken through a special silencer within the casing, so that the tools are virtually noiseless in operation.

In addition to the assembly stations at which several nuts are tightened simultaneously, there are, of course, many more stations at which nuts are tightened singly. Once again, SR.50 motors are used, and in greater or lesser degree each tool is designed for one specific operation. Of the single spindle tools, the most interesting are probably those designated type SR10/35. This is a newly developed worm type spanner that is very powerful. The motor drives the nut runner through a worm gear reduction unit which is absolutely irreversible. When the nut is run down by power, the operator can, while the air is fully switched on, use the handle of the tool as a ratchet spanner that is completely free from backlash. As a result, this type of tool can be used for the final tightening of a nut with a minimum of handle movement even in the most confined space on a chassis. This tool is, in fact, generally used for tightening nuts which are relatively inaccessible. Two applications of this newly developed tool are illustrated in Figs. 6 and 7. One of these spanners in use for tightening the rear bumper bracket is shown in Fig. 6 and one with a long reach extension for fixing the radiator support bracket is shown in Fig. 7.

As the assembly is carried out on a moving conveyor, it is important that all the tools come readily to hand when required. Therefore, special attention has been paid to positioning the tools to give optimum operating convenience and efficiency. When not in use many single spindle tools are lodged in special rests at the side of the conveyor. Other tools may be carried by individual suspension units, while yet others are carried on overhead arms that radiate horizontally from a central hub, see Fig. 8. These arms are free to rotate so that any of the tools can be brought quickly and easily into the working position.

Chemical Finishing

THE 1951 Ministry of Supply and Board of Trade Orders (Nos. 1048 and 1049 respectively), prohibiting the use of nickel for plating purposes, leave motor manufacturers with lacquer, paint and chemical finishes as alternatives to the usual nickel or chromium plating. A chemical process developed by Metal Processes Ltd., Kingsbury Road, Erdington, Birmingham, 24, is claimed to be suitable for a variety of vehicle fittings such as lamps, hub discs, filler caps and gear levers.

An article of copper, brass, or steel previously coppered or brassed, if immersed in a heated solution of "Niklit" powder in ordinary tap water, receives a finish simulating that of nickel in five to ten minutes, according to size. Neither electrical equipment nor skilled labour is required, and there is a substantial saving in cost. The initial surface finish of the metal is unaffected: a polished surface will not require re-polishing after treatment.

As the process is chemical it causes no dimensional change and the finish cannot flake. It is claimed it will not tarnish over a period of years.

MACHINABILITY

A Résumé of American Investigations of Methods of Increasing Production and Reducing Costs

OPTIMUM output, that is the best possible combination of workpieces produced and tools expended, is at any time of primary importance in production economics. To-day, with the small tool situation becoming ever more serious, this matter is of even greater importance, It has been the subject of detailed study in the United States of America, and these notes are based on the results of recent investigations in that country. The original impetus for the investigations arose from the difficulties encountered in machining the high-temperature alloys used in turbo-jet aircraft engines. Of more immediate interest to the automobile industry is the work that has been carried out on cast irons.

In the past thirty years there have been very great advances in the application of machining processes. New tool materials have been developed to give much higher rates of metal removal. To keep pace with these materials, machine tools have been designed to employ faster cutting speeds. By contrast surprisingly little has been done to apply the available metallurgical knowledge to the production of materials with better machinability properties. In fact, until very recently, specific treatment and control of material to give the best possible degree of machinability has been seriously neglected. Certainly, production conditions have not always been adjusted to take full advantage of existing metallurgical knowledge. For example, it has been definitely established that slight variations in microstructure can be responsible for as much as a five-fold change in tool life at a given speed, yet little attention is paid to the problem of controlling the microstructure. Far closer scrutiny of microstructure is essential if optimum production results are to be obtained. Furthermore, once the desirable microstructure has been established, extremely close control should be exercised to maintain it.

Cast irons

In terms of both its mechanical and application properties, cast iron has long been taken for granted. Yet, cast iron is a term applicable to a large range of alloys from very soft and crumbly material to glass hard and brittle, and even to hard, strong and ductile materials with mechanical properties comparable with those of some steels. The investigations on which these notes are based, have disclosed many interesting points about the machinability of cast irons. Of these, probably the most important is the influence of microstructure.

The hardness and strength of a metal, as well as its properties in machining operations, are known to depend upon the microstructure. This structure in turn depends upon the chemical composition and heat treatment. In a cast iron the microstructure is governed in great measure by the cooling rate, which is influenced by the pouring temperature, the moisture in the sand mould, the wall thickness or section size in the casting, and the atmospheric conditions. A single casting may include structures ranging from ferrite to white iron because of uncontrolled cooling attributable to the foregoing factors. Test bars cast at the same time to provide specimens for mechanical testing may show structures and properties that are not at all representative of the actual casting.

The microstructures of cast irons are not limited to specific chemical compositions, and any one structure may be produced in a variety of ways. It has also been established that irons of the same microstructure, regardless of chemical composition, section size, cooling rate or mechanical properties, have approximately equal machining properties. While mechanical properties and machining properties both depend upon microstructure, they are not necessarily dependent upon each other. For example, Brinell hardness is often used as a rough guide to machinability, but it can lead to completely false results. It has also been established that certain changes in microstructure lead to longer tool life with higher cutting speeds, and at the same time, contrary to expectation, they also produce stronger and harder materials. In fact, microstructure is the only reliable guide to machinability. In this connection, it must be stressed that it is necessary to determine the structure in the immediate area to be machined, since structure will vary throughout the casting.

TABLE I. HARDNESS OF CAST IRON MICROSTRUCTURES AT ROOM TEMPERATURE

Microconstituent	Knoop microhardness 100 gram load	
	Range of values	Average value
Free graphite	15–40	20
Free ferrite	215–270	240
Pearlite	300–390	350
Acicular	400–495	445
Steadite	600–1200	950
Free carbide	1000–2300	1500

Cast iron microstructures

As the microstructure of a cast iron is all important for machinability, brief reference may be made to the various micro-constituents usually associated with the cast irons. Common grey cast irons have a structure composed of pearlite and flaky graphite. These may be either coarse or fine. Generally, coarse pearlite is associated with coarse graphite and fine pearlite with fine graphite, since the factors which produce the coarseness in one also produce it in the other. Fine pearlite is associated with a rapid cooling rate, low carbon content, low silicon content, and the presence of alloying elements such as chromium, vanadium and molybdenum. Coarse pearlite develops when the cooling rate is low and when the iron is not highly alloyed. Grey cast irons may also contain ferrite, steadite, free carbide, manganese sulphide and various inclusions.

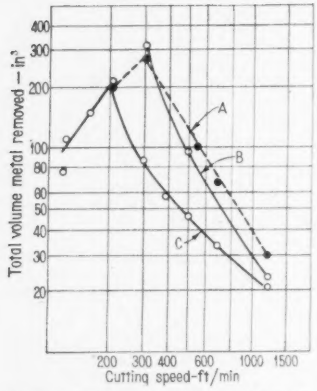
Ferrite. In the as-cast condition, free ferrite may be present in quantities up to about 10 per cent. To obtain a very large proportion of free ferrite, 50 per cent or more, the cast iron must be annealed for periods of from one to eight hours at temperatures from 1400-1600 deg F.

Steadite. This is a eutectic composition of iron, carbon and phosphorus. It is commonly found in irons containing more than 0·10 per cent phosphorus. With the ordinary cast irons, steadite is present in quantities less than five per cent. Its presence in such amount has little effect on tool life.

Free carbide. This constituent occurs in irons that have been cooled very rapidly. The formation of free carbide is accelerated by such alloys a chromium and vanadium. Even small amounts of free carbides are extremely detrimental to tool life.

Acicular structure. An acicular structure in cast iron can be produced by the addition of chromium, nickel and molybdenum in sufficient quantities. This structure gives an iron that is hard and strong and difficult to machine.

White iron. White iron may form in small areas in thin sections that have



A medium pearlite graphite with no steadite or free carbide
B medium pearlite graphite with 5 per cent steadite
C medium pearlite graphite with 5 per cent free carbide

Fig. 1. The effect of steadite and free carbides on machinability

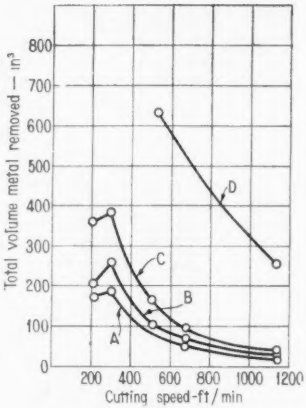
been cooled very rapidly. Casting flash often contains white iron. This structure is about 50 per cent pearlite and 50 per cent free carbide. Machining operations should not be carried out in areas where this structure is likely to form.

Inclusions. Manganese sulphide occurs as a trace in cast irons. Its presence would be beneficial to machinability, but the amount present is usually so little as to be of no consequence. Sand and slag may also occur as the result of faulty foundry control.

Malleable iron. Malleable iron contains clusters or nodules of free graphite instead of flake graphite. The matrix structure can be similar to that of grey cast iron, that is, it may contain pearlite, free ferrite, steadite and manganese sulphide. The nodular graphite produces higher mechanical strength than does flake graphite. Malleable iron is produced by heat treating cast irons for from one to five days. Its use is limited to small section sizes and it is more expensive than cast iron. Recently, nodular irons have been produced by direct casting through the addition of magnesium or cerium alloys in the ladle. Such irons contain a nodular graphite in a matrix of pearlite or free ferrite. These as-cast nodular irons are not limited in size. They have mechanical properties similar to those of malleable iron.

The hardness of the micro constituents determines the abrasiveness of the structures, a most significant factor affecting tool life. In general, the harder or more abrasive constituents give a shorter tool life for a given set of machining conditions. The microhardness at room temperature of the constituents of cast iron, as measured with a Tukon micro-hardness testing machine, are given in Table I.

Free graphite is soft and amorphous. It acts as a free machining agent. Free ferrite is the softest major constituent in cast irons. Pearlite is about 50 per cent harder than free ferrite and thus gives a shorter tool life. The presence of steadite, which is very hard at room temperature, is not particularly detrimental to machinability when the content is less than 10 per cent. This may be attributed to the low melting point of steadite, which could make it much softer at the cutting temperature. Free carbide is harder than tool steel and is almost as hard as sintered carbide tool material. As a result, small amounts of free carbide, even as little as three to five per cent, drastically reduce tool life. The effect of free carbide is illustrated in Fig. 1, while Table II shows the relative machinability of eight typical flake graphite cast irons. The data in this table are for carbide cutters and for cutting speeds in excess of 300 f.p.m.



A fine pearlite-graphite, 218 BHN
B medium pearlite-graphite, 215 BHN
C coarse pearlite-graphite, 195 BHN
D ferrite-graphite, 121 BHN

Fig. 2. Tool life for various flake graphite structures

in a 12 in circle centrally located in relation to the work. Cast iron test blocks, 6 in wide and 20 in long were used.

The tests were run at cutting speeds up to 1,200 f.p.m. and with feeds per tooth up to 0·060 in. A carbide-tipped face mill was used in all tests. It has 3 deg axial rake, 3 deg radial rake, 30 deg corner angle and a resultant rake of 4 deg. The depth of cut was 0·187 in and the feed per tooth 0·015 in. Generally, the tool life is based upon the volume of metal removed for 0·030 in tool wear, but certain tests were based on 0·015 in and 0·005 in tool wear.

Selection of the cutting speed and the feed for any specific job is of major importance. Once the microstructure to be machined has been established, the optimum cutting speed should be determined to suit the nature of the job. The example, short run batches should normally be machined without there being necessity to re-sharpen the cutter. For such classes of work the best procedure is to calculate the total volume of metal to be removed and

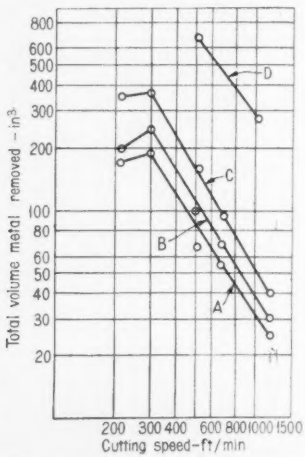


Fig. 3. Tool life, plotted on log-log scale, for the same cast irons as in Fig. 2

TABLE II. TOOL LIFE INDEX NUMBERS FOR TYPICAL FLAKE-GRAPHITE CAST IRON MICROSTRUCTURES

Structure	Brinell hardness	Index number
Ferrite, graphite (Full anneal)	120	20
50 per cent ferrite, 50 per cent graphite (Partial anneal)	150	10
Coarse pearlite, graphite	195	2
Medium pearlite, graphite	215	1·5
Fine pearlite, graphite	218	1
Fine pearlite, 5 per cent Steadite graphite	197	1
Fine pearlite, 5 per cent free carbide, graphite	240	0·3
White iron	over 500	

Test results

Face milling tests have been carried out to establish data on tool life, surface finish and power for as-cast and annealed cast irons in several microstructural variations. Testing was carried out on a fixed bed milling machine with a single-tooth cutter held in a 500 lb flywheel. The tooth swung

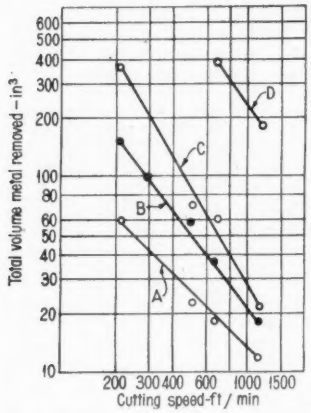


Fig. 4. Tool life for 0·015 in flank wear in cutting the same cast irons as in Fig. 2

divide by the number of teeth in the cutter to obtain the tool life required per tooth. Large quantity, continuous production jobs require a tool life such that cutter changes will be economically synchronized with the flow of materials and the schedule of operations.

Test results for four major cast iron structures are shown in Figs. 2 and 3. In Fig. 2 the results are plotted on rectangular co-ordinates, and the same results plotted on log-log co-ordinates are shown in Fig. 3. They show that the maximum tool life is obtained at a cutting speed of 300 f.p.m. for all four structures. Below this speed, tool wear is not uniform and localized wear shortens tool life. Maximum tool life is not however the sole consideration. Rather, the ultimate aim is to get the highest possible production at the lowest possible price. The data presented in Figs. 2 and 3 can help the selection of the most economical speed and feed.

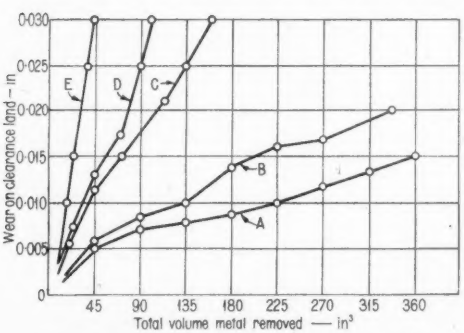
In using these data it is first necessary to determine the type of microstructure that is to be machined. This involves the economics of annealing and heat treating. For example, when final mechanical characteristics are not critical, the cost of annealing may be justified by the higher production rate and the increased tool life it makes possible. At a constant tool life of 200 cubic inches of metal removal the increase in production rate as between a fine pearlite structure and fully-annealed ferritic iron is 500 per cent, that is, the cutting speed can be increased from 300 f.p.m. to 1,500 f.p.m. or, at a fixed cutting speed of 500 f.p.m. the increase in tool life would be 800 per cent, from 85 to about 680 cubic inches of metal removed.

In continuous high quantity production, tool life should be so determined that tool changes are synchronized with other necessary production breaks. For such applications, the cutter, as well as the speed and feed, is chosen so that all the factors are in equilibrium. Since the tool life and point will seldom be exactly 0·030 in, because of the many variables involved, Figs. 4 and 5 are given to show metal removal for 0·015 in and 0·005 in tool wear. Fig. 6 gives the rate of tooth wear at different cutting speeds, while Fig. 7 shows the actual time the tooth is in contact with the work, which is something less than one-half cutter revolution. It is not the total time required to remove the volume of metal. To convert this into information pertinent to a specific job, the ratio of cutting time to total time must be calculated from the cutter diameter, the width of the cut and the location of the cutter with respect to the work.

Surface finish and power

For a given cast iron and cutter, surface finish improves as cutting speed increases. Between 22 f.p.m. and 1,200 f.p.m., the Profilometer reading falls by from 50 to 100 microinches, depending upon the microstructure of the cast iron and the condition of the cutter. At constant cutting speed, the surface finish is better with irons of finer structure, especially as the quantity of free graphite decreases. If feed marks are undesirable, the face or end cutting angle may be reduced to zero to give a land for at least twice the feed per tooth. This will further improve the finish by about 50 microinches.

Machining power, that is, the horse-power needed to remove metal at a given rate, is about the same for all pearlitic cast irons, but is considerably less for ferritic iron. Unit power decreases as feed is increased, because a thicker chip is removed from the same contact area between tool and chip. The power required may increase as much as three times when the cutter becomes dull.



A 210 f.p.m.; B 250 f.p.m.; C 510 f.p.m.; D 670 f.p.m.; E 1,150 f.p.m.

Fig. 6. Rate of tooth wear at different cutting speeds

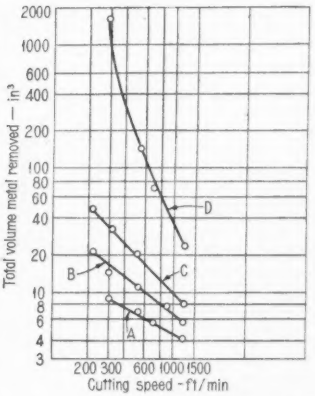
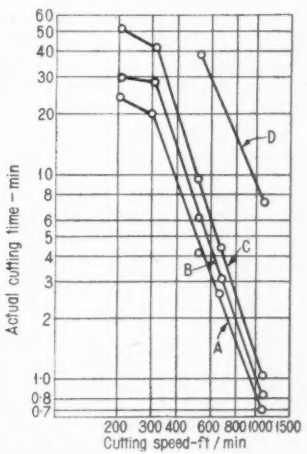


Fig. 5. Tool life for 0·005 in flank wear in cutting the same cast irons as in Fig. 2

Comparison of flake and nodular cast irons

Investigations have been carried out for comparing the tool life, the surface finish and the power requirements in turning operations on grey cast irons (flake graphite) and ductile cast iron (nodular graphite). In these tests the emphasis was on the tool life obtainable with material of a certain microstructure, and on the mechanical properties, hardness, strength, etc, of the structure.

Four compositions of flake graphite irons and three of nodular irons were used. The chemical analyses and the mechanical properties of these irons are



A fine pearlite-graphite, 218 BHN
B medium pearlite-graphite, 215 BHN
C coarse pearlite-graphite, 195 BHN
D ferrite-graphite, 121 BHN

Fig. 7. Tool life in actual cutting time

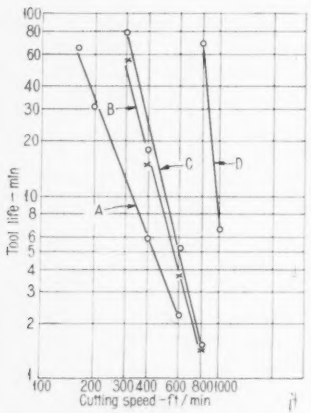
TABLE III. COMPARATIVE COMPOSITIONS, MECHANICAL PROPERTIES AND TOOL LIFE OF NODULAR-GRAPHITE AND FLAKE-GRAPHITE CAST IRONS

Composition	Nodular Graphite Irons					Flake Graphite Irons			
	2 per cent as cast	4 per cent as cast	17 per cent as cast	20 per cent annealed	22 per cent annealed	Acicular Iron	Fine Pearlite	Coarse Pearlite	Ferrite Annealed
Carbon	3·33	3·41	3·82	2·79	3·69	2·78	3·22	3·17	3·47
Manganese	0·45	0·42	0·23	0·53	0·27	1·03	0·62	1·10	0·35
Phosphorus	0·11	0·09	0·082	0·09	0·045	0·12	0·08	0·10	0·05
Sulphur	0·018	0·014	0·018	0·014	0·014	0·064	0·097	0·108	0·133
Silicon	2·66	2·82	3·04	2·76	2·64	2·30	2·10	2·44	1·62
Nickel	1·65	0·81	1·13	1·59	1·15	2·11	0·85	—	—
Magnesium	0·079	0·073	0·082	0·071	0·060	—	—	—	—
Molybdenum	—	—	—	—	—	0·32	—	—	—
Chromium	—	—	—	—	—	—	0·64	—	—
Mechanical properties									
Brinell hardness	265	215	207	183	170	263	225	195	100
Ultimate tensile strength lb/in²	97,250	93,000	84,700	77,000	70,000	59,000	45,000	35,000	17,500
Yield point lb/in²	79,000	72,000	69,800	62,000	56,000	—	—	—	—
per cent elongation 2 in	2·0	4·0	17·5	20·0	22·0	—	—	—	—
per cent reduction in area	1·5	3·5	16·5	21·0	22·0	—	—	—	—
Cutting speed ft/min for 280 in² tool life	200	310	420	600	970	150	310	325	960

given in Table III. All seven irons were machined in the as-cast condition. Subsequently, one of the grey irons and two of the nodular irons were annealed and again machined. In every case the annealing cycle was:—hold at 1,650 deg F for 30 minutes, furnace cool to 1,275 deg F and hold for five hours, then air cool.

In the flake graphite irons the microstructures investigated were essentially: acicular, fine pearlite, coarse pearlite and ferrite. The acicular structure is typical of a hard, highly alloyed cast iron. Ordinary as-cast grey iron is represented by the fine and coarse pearlite, while the ferritic iron is the same iron annealed. The respective Brinell hardnesses are 263, 225, 195 and 100.

From their respective ductilities, the three nodular irons are designated the 2 per cent, the 4 per cent and the 17 per cent analyses. The 2 per cent has a structure consisting of about 80 per cent fine pearlite and 20 per cent free ferrite plus nodular graphite, with the ferrite surrounding the graphite nodules. In the four per cent iron the composition is about 40 per cent fine pearlite and 60 per cent ferrite plus nodular graphite, and the 17 per cent structure is similar. The respective Brinell hardnesses as-cast are: 275, 215 and 207. Annealing to produce ferritic structures was carried out on the 2 per cent and the 17 per cent irons. As a result of the annealing the elongation of the 2 per cent increased to 20 per cent, and that of the 17 per cent to 22 per cent. The respective hardnesses were 183 and 170.



A acicular iron, 263 BHN
B fine pearlite, 225 BHN
C coarse pearlite, 195 BHN
D ferrite, 100 BHN

Fig. 8. Tool life in minutes in cutting various flake graphite cast irons

Test procedure

Standard carbide-tipped tools were used in a special tool-holder that gave maximum rigidity and eliminated overhang. Tool grinding was carried out on a Cincinnati tool grinder with a resinoid-bonded diamond wheel. In re-sharpening, all previous defects plus 0·005 in were removed from all surfaces of the carbide. The tool angles were:—zero back rake, 6 deg side rake, zero side cutting-edge angle, 6 deg end cutting-edge angle, 6 deg relief, and 0·015 in nose radius. In every test the depth of cut was 0·100 in and the feed 0·010 in per revolution. Cutting speeds from 150 to 1,000 f.p.m. were employed. All tests were run without a cutting fluid, and 0·030 in flank wear was chosen as tool life end point.

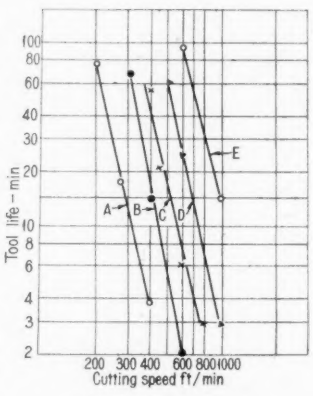
The relationships between tool life and cutting speed for the several irons are shown in Figs. 8 and 9. Tool life in minutes against cutting speed for flake graphite is shown in Fig. 8 and for nodular graphite in Fig. 9. Tool life in cubic inches of metal removal is shown in Fig. 10. Of the flake graphite irons, the acicular is the least machinable and the ferritic the easiest to machine. For 200 cubic inches metal removal at 0·030 in flank wear, the following cutting speeds could be used for the flake graphite irons:—

Acicular .. .. .. 150 f.p.m.
Fine pearlite, graphite .. 310 f.p.m.
Coarse pearlite, graphite .. 325 f.p.m.
Ferrite, graphite .. .. 960 f.p.m.

With the nodular-graphite irons, tool life improved as the ductility increased and as the hardness decreased. For 200 cubic inches metal removal at 0·030 in flank wear, the following cutting speeds could be employed:—

2 per cent as-cast .. .. 200 f.p.m.
4 per cent as-cast .. 310 f.p.m.
17 per cent as-cast .. 420 f.p.m.
20 per cent annealed .. 600 f.p.m.
22 per cent annealed .. 970 f.p.m.

These results are informative. The acicular structure gives the hardest and strongest of the grey iron series and a cutting speed of 150 f.p.m. can be

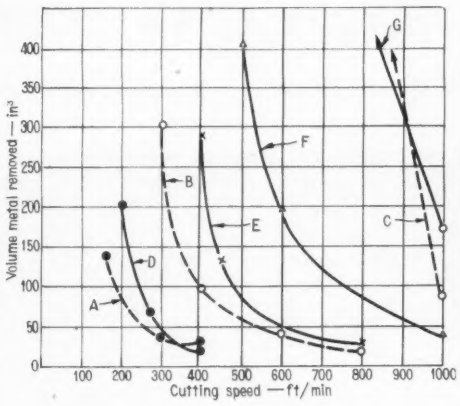


A 2 per cent as-cast, 265 BHN
B 4 per cent as-cast, 215 BHN
C 17 per cent as-cast, 207 BHN
D 20 per cent annealed, 183 BHN
E 22 per cent annealed, 170 BHN

Fig. 9. Tool life in minutes in cutting various nodular graphite cast irons

used for 200 cubic inches metal removal, whereas the weakest and softest nodular iron, the 22 per cent annealed, is much stronger than the acicular iron and a cutting speed of 970 f.p.m. can be used for a tool life of 200 cubic inches metal removal. In a flake-graphite iron, this speed is possible only when the material is in the fully-annealed condition with only one-fourth the strength of acicular iron. The strongest nodular iron equals the acicular in hardness, is 50 per cent stronger and for the same tool life permits the use of a one-third higher cutting speed.

Surface finish is affected by microstructure, tool sharpness and cutting speed. Higher cutting speeds and worn tools both tend to improve surface finish on both flake-graphite and nodular-graphite cast irons. Test results are shown in Figs. 11 and 12. In general, the surface obtained in machining the nodular graphite irons is of the same order as that obtained on the flake-graphite irons, but the 2 per cent as-cast nodular-graphite iron machined with a far higher degree of finish, even at low cutting speeds, than any of the other nodular- or flake-graphite irons. For a given set of conditions, annealed cast irons show the poorest surface finish. This is because the structure has more ferrite and is coarser.



A, B and C as in Fig. 8 D, E, F and G as in Fig. 9

Fig. 10. Comparative tool life in volume of metal removed.

Drilling tests

Multi-spindle drilling is one of the commonest operations in an automobile factory. In connection with this type of operation, an interesting series of tests was carried out to determine two important points, (a) to discover if any improvements could be made in drill geometry to increase drill life, and (b) to determine statistically the optimum production point at which to remove and replace all drills in order to minimize idle time and work stoppages.

Five tests were carried out on a 16-spindle machine set at 0·008 in feed per revolution and a spindle speed of 700 r.p.m. to give a cutting speed of 85 f.p.m. All drills were of 18-4-1 high speed steel and the drill particulars were :—

Test 1. Hand ground, 118 deg point angle.

Test 2. Hand ground, 100 deg point angle.

Test 3. Machine ground, 100 deg point angle.

Test 4. Machine ground, 100 deg point angle with an additional 60 deg chamfer on the drill lip.

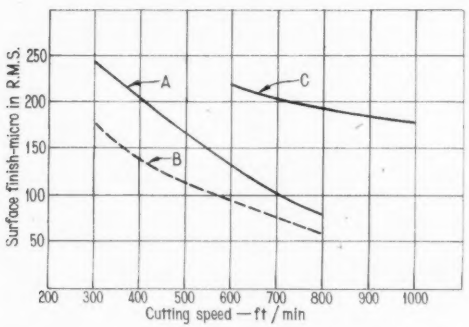
Test 5. Similar to 4, but the casting skin was removed by milling before the drilling operation.

In production of this character, drills are usually run until they fail, because there has been no practical way to measure the degree of wear on individual drills. Drill failure is readily detected, by noise, chatter and over-heating and by the damaged holes that are produced. In test No. 1 the first drill failed after 75 holes, the second at approximately 220 holes, while by 360 holes 10 drills had failed. Two drills were still in service after 500 holes.

In test No. 2 the early failures were higher than in test No. 1, but after 650 holes there had been only eight failures and the remaining eight drills were still in service. In test No. 3, the first with machine ground drills, only six drills had failed after 600 castings had been machined. With the double-point, machine ground drill used in test No. 4 there were failures for the first 630 holes per drill. At 800 holes, three had failed, at 900 seven, at 1,000 fourteen, and two drills lasted for 1,100 holes. Test No. 5, with the same form of drill, but with the work milled to remove the casting skin, produced 1,100 holes per drill with only two failures.

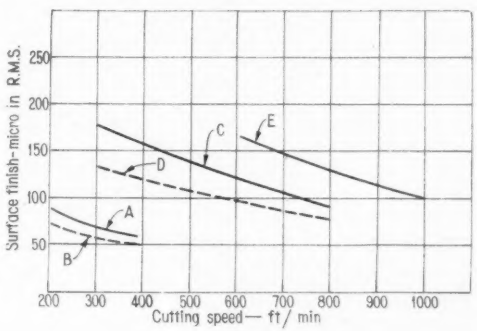
Microstructure was not one of the controlled variables in these tests, but all the castings were assumed to have the same structure. There is, however no doubt that structure variations in small areas were responsible for most of the early drill failures. The presence of carbides and white iron, constituents that frequently occur in thin sections or near bosses, can be extremely detrimental to tool life. In this connection, the milling of the casting skin prior to drilling, in addition to removing scale and inclusions, often gets far enough below the surface to enter a region with a microstructure different from that at the surface.

As a result of these tests, it was decided to standardize on the production sequence and type of drills used in test No. 5, and to remove and replace all drills after every 800 castings.



A pearlite-graphite, dull tool
B pearlite-graphite, sharp tool
C ferrite-graphite, sharp tool

Fig. 11. Surface finish on flake graphite cast irons



A 2 per cent as-cast, sharp tool
B 2 per cent as-cast, dull tool
C 4 per cent as-cast, sharp tool
D 4 per cent as-cast, dull tool
E 20 and 22 per cent annealed, sharp tool

Fig. 12. Surface finish on nodular graphite cast iron

GRINDING VALVE TAPPETS

A Special Scrivener Machine for End-forming Operations

SOME of the latest developments in valve tappets have raised difficulties in grinding the head forms. These head forms may be a plain flat, a chamfer-and-flat, or a slight dome, but in every case the head form must be square with the axis of the tappet within plus or minus one degree. For grinding a range of such tappets an interesting automatic, special-purpose machine has been developed by Arthur Scrivener Limited, Tyburn Rd., Birmingham. It is illustrated in Fig. 1.

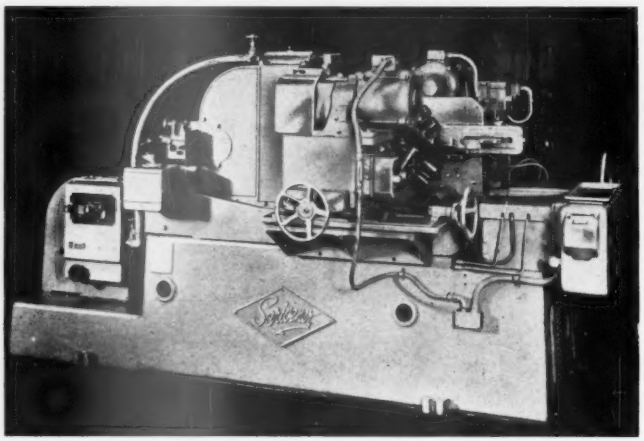


Fig. 1. Special Scrivener machine for grinding valve tappet heads

This machine may be considered as being comprised of four individual units. They are: (1) the grinding head, (2) a five-station turret, (3) an automatic loader, and (4) the controlled-cycle mechanism. The grinding wheel is 20 in diameter by 2 in wide. It runs at 1,150 r.p.m. Wheel-truing to the specified form is effected by means of an hydraulically-operated form-truing apparatus working in conjunction with a suitable former plate. While it is in contact with the grinding wheel, the workpiece is rotated by an independently driven control wheel 10 in diameter and 1¾ in wide.

Fig. 2 shows a detail view of the workpiece in contact with the grinding wheel and Fig. 3 a detail view of the loading chute, turret and control wheel. In operation the workpieces W are fed down an inclined chute, and loaded on to the pins of a five-face turret by the pusher A as the turret makes a momentary pause at this loading station. The turret is indexed by a conventional Geneva mechanism that indexes 10 times for one complete revolution. This is necessary to give an intermediate station for loading.

When the turret has been indexed to the grinding position, the controlled-cycle mechanism at the side of the machine operates to advance the turret to bring the work into contact with the grinding wheel B. The control wheel C imparts rotation to the work. As it is essential that the ground head of the tappet should be square with the axis, the work is located from the previously ground body, and the turret pin upon which it is mounted is merely a carrier. During the actual grinding operation, the pressure of the control wheel forces the tappet against two parallel rollers R alongside the carrying pin. This gives three-point location and ensures that the head is ground with the sides within the specified tolerance. The carrier pins are fixed in hardened and ground steel blocks on the turret faces.

Automatic loading from the inclined chute is effected by the pusher A which is operated by a linkage and cam from the turret head so that the movements of the turret and the pusher are synchronized. On the valve tappets for which this machine is used, the stock removal is usually in the order of 0·008 in to 0·010 in. Production is at the rate of 300 pieces per hour.

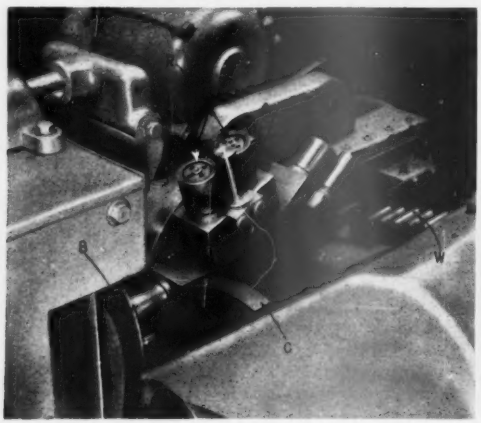


Fig. 2. Detail view of work applied to grinding wheel

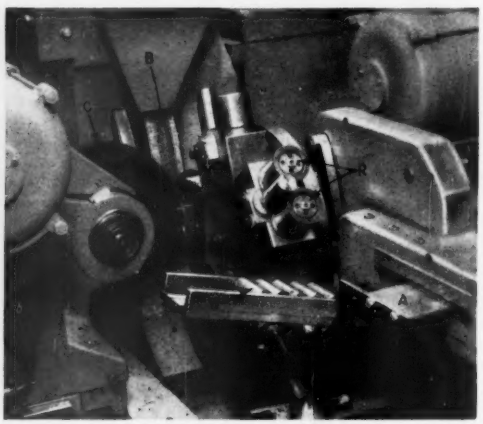


Fig. 3. Loading chute, turret and control wheel

CONSERVING SCARCE MATERIALS

New Specifications for Casehardening Steels

SHORTAGES of materials have been causing increasing concern to a number of British industries and specifically to motor manufacturers. To a considerable extent this is a consequence of the decision of the Western Powers to carry out a vast programme of rearmament. Efforts have been made to produce sufficient goods and services to satisfy not only the requirements of rearmament but also the demands of civilian consumption, which has been rising steadily in the post-war years. An additional factor accentuating the situation is that world production of basic raw materials has not in recent years been increased at the same rate as world production of goods. Much is being done to expand supplies of basic materials in different parts of the world, but these measures are mainly long term in character and cannot be expected to relieve the current scarcities resulting from rearmament. The position is further aggravated by the need for stockpiling to cover the potential requirements of full-scale rearmament.

During June and July last year a team of specialists visited the United States of America to study the steps taken in that country to meet these shortages and to report on any measures that might be of help to British industry. The team confined its attention to the engineering and allied industries and as a result the survey is almost wholly concerned with scarcity in supply and economy in the use of metals. A large section of the Report* deals with advances that have been made in recent years in the use of low-alloy or plain carbon steels.

British industry, of course, has had a good deal of experience, both during and since the War, in effecting economies in scarce materials. The decision to send a team to America was no reflection on the careful husbandry exercised in Britain, but was an expression of the determination to investigate all means that could possibly assist in conserving scarce materials, many of which originate in the U.S.A.

The problems arising from the serious shortages of alloying elements, particularly nickel, have necessitated reconsideration of many alloy steel specifications. Alloy casehardening steels have traditionally been rich in nickel, although it has been realized for some years that this element is not strictly necessary in amounts greater than approximately 2 per cent, even for the highest core strengths. Covering the tensile strength range of from 40–85 tons/in², B.S.970 originally specified the En 33 to En 39 series of low-carbon, alloy casehardening steels in which the nickel content varied between 2 per cent and 5 per cent. The straight 5 per cent Ni steels En 37 and 38 were suitable for core strengths up to 65 tons/in² minimum, and were accompanied by other steels of lower nickel content, usually containing chromium with or without molybdenum (En 33 to En 36). For the 85 tons/in² range, a 4·25 per cent Ni, 1·25 per cent Cr steel, with or without molybdenum, was thought to be necessary.

Faced by acute nickel shortages during the war, the Germans evolved a 2 per cent Ni, 2 per cent Cr-Mo steel, later incorporated in B.S.970 as En 320. It was soon discovered, however, that the chromium content of this steel was too great, and although the minimum core strength of 85 tons/in² was attainable, the carburized case was of poor quality owing to the formation of an excessively heavy network of free carbides. A further attempt at producing a lower alloy casehardenning steel was En 325, containing about 1·75 per cent Ni, 0·5 per cent Cr, and 0·25 per cent Mo. While this steel was quite successful, the core strength achieved on blank-carburized $1\frac{1}{8}$ in diameter test bars after the standard refining and hardening treatments was only 55 tons in².

Reconsideration of specifications was entrusted to a Technical Committee, with Mr. H. H. Burton as its chairman, which is responsible to the Ministry of Supply Steel (Rearmament) Panel, under the chairmanship of Mr. E. W. Senior. One of the earliest tasks of this committee has been to draw up emergency specifications replacing the alloy casehardening steels En 33 to 39 and En 325. These new specifications, En 351 to En 355 and En 361 to En 363, have been endorsed by the appropriate Committees of the British Standards Institution, and are now issued as an Addendum to B.S.970†.

The compositions and properties of the new steels are summarized in the table. Apart from the large saving in nickel, the outstanding features are as follows:—

(1) Manganese content has been slightly raised where possible.
(2) Chromium content has been regulated with due regard to the nickel content, to avoid excessive free carbides in the case.
(3) Full advantage has been taken of the disproportionate beneficial effects of low molybdenum contents. Most of the molybdenum in all the steels, except En 354 and En 355, may be expected to

*Saving Scarce Materials. Anglo-American Council on Productivity, 21, Tothill Street, London, S.W.1. Price 2s. net.

†Addendum No. 1: November, 1951, to B.S.40: 1947, Wrought Steels, *Emergency Specifications for Case-Hardening Steels*. British Standards Institution. Price 1s. net.

COMPOSITIONS AND PROPERTIES

	En 351	En 352	En 353	En 354	En 355	En 361	En 362	En 363
Tensile Range	45-55	55-65	65-75	75-85	85 min	45-55	55-65	65-75
En Steels Replaced	33,34,37	35,36,325	36,38	New Range	39	33,34,37	35,36,325	36,38
Composition—per cent								
Carbon	0·20 max	0·22 max	0·22 max	0·22 max	0·20 max	0·13-0·17	0·18-0·23	0·22-0·26
Silicon	0·35 max	0·35 max	0·35 max	0·35 max	0·35 max	0·35 max	0·35 max	0·35 max
Manganese	0·60-1·00	0·50-1·00	0·50-1·00	0·50-1·00	0·40-0·70	0·70-1·00	0·70-1·00	0·70-1·00
Nickel	0·60-1·00	0·85-1·25	1·00-1·50	1·50-2·00	1·80-2·20	0·40-0·70	0·40-0·70	0·40-0·70
Chromium	0·40-0·80	0·60-1·00	0·75-1·25	0·75-1·25	1·40-1·80	0·55-0·80	0·55-0·80	0·55-0·80
Molybdenum	0·10 max	0·10 max	0·08-0·15	0·10-0·20	0·15-0·25	0·08-0·15	0·08-0·15	0·08-0·15
Sulphur	0·05 max	0·05 max	0·05 max	0·05 max	0·05 max	0·05 max	0·05 max	0·05 max
Phosphorus	0·05 max	0·05 max	0·05 max	0·05 max	0·05 max	0·05 max	0·05 max	0·05 max
Ultimate Tensile strength—tons/in²	45 min	55 min	65 min	75 min	85 min	45 min	55 min	65 min
Elongation—per cent	18 min	15 min	12 min	12 min	12 min	18 min	15 min	—
Izod Impact—ft-lb	30 min	20 min	20 min	20 min	25 min	25 min	15 min	—

come from remelted alloy scrap, at least during the change-over period.

(4) The gap between the 65- and 85-ton steels has been closed by the provision of En 354, suitable for a tensile strength of 75 tons in^2 minimum.

(5) Since not only molybdenum but also the other alloying elements will be obtained as far as possibe by remelting alloy scrap, two alternative specificatons have been provided for each of the three tensile strength ranges between 45 and 65 tons in^2.

From the user's point of view the chemical composition of a steel is of secondary importance to its suitability for the task it is called upon to perform. The new steels will caseharden satisfactorily by all the normally used carburizing processes and will give the required minimum ultimate tensile strength. That the hardenability is adequate is evident from the fact that oil hardening is specified. The only apparent practical difference between these and the original En alloy casehardening steels is that they are lower in Izod impact value. It must, of course, be pointed out that, as is usually the case, the actual impact values obtained from correctly made and processed steel are considerably in excess of the specified minima.

In terms of actual behaviour, the difference between a steel with 50 ft-lb Izod and another steel with 25 ft-lb rating but otherwise identical properties, is difficult to define. For all normal purposes it is probably non-existent. The real value of Izod testing is as an indication to the producer or user that one particular batch is comparable in quality with another batch of the same material. Where the datum line is located is a matter of much less concern, provided, of course, that highly notch-sensitive material is not put into service under adverse conditions.

With casehardening steels, the question is further complicated by the known fact that the carburized case itself has poor Izod properties. So long as the core has adequate strength and ductility, therefore, its impact value would appear to be a secondary consideration.

It must be remembered that the new alloy casehardening steel specifications are the end result of very extensive investigations in which the steelmakers and the major steel users have co-operated wholeheartedly. Those responsible for the work are to be congratulated on their thoroughness and on the scale of the economies they have achieved.

Some comfort may be derived from the statement of Dr. J. F. Thompson, Chairman and President of the International Nickel Company of Canada Ltd., on 20th December, 1951. He announced that world production of nickel, excluding Soviet Russia and her satellite countries for which no reliable estimates are available, for the year 1951 would be approximately 295 million pounds. This represents a 10 per cent increase over production in 1950. Of this figure Canadian producers were responsible for 275 million pounds, or more than 90 per cent. Extension of capacity by established producers, as well as new potential producers, will ensure continuance of supply and by 1954 is conservatively estimated to provide an increase of approximately 30 per cent over that available before the outbreak of the Korean conflict.

ELECTRICAL MACHINING

A Novel Method of Metal Removal

A NEW machining method which employs the direct use of electric energy, has been developed by the Method X Corporation of Philadelphia, Pennsylvania, U.S.A., an affiliate of the Firth Sterling Steel and Carbide Corporation. It can be used for machining any electrically conductive material. The Method X machine, as it is called, removes metal by means of an electric spark discharge that does not otherwise affect the physical or chemical characteristics of the work material.

The machining action arises from a mechanical, not a thermal, effect of electricity. This effect is obtained by the use of extremely high current densities, which set up internal mechanical stresses and thereby cause the metal particles to detach themselves from the work material without resort to melting. As a result, the exposed surfaces are unchanged. Some remarkable results have been achieved. For example, lapping of less than 0·001 in will produce any desired finish down to 0·15 microinch on sintered carbide. Centre-to-centre spacing of holes can be controlled to approximately 0·0005 in and blind hole fillet radii can be produced as small as 0·002 in. This new process is particularly suitable for use on such hard-to-machine materials as sintered carbides, super-alloys, S-816, vitalium, and hardened steels. Such materials can be machined economically and rapidly into complicated geometric shapes.

An electrode made to the desired shape is fed directly into the workpiece by the Method X machine to produce the required form. This produces a rough shaped hole. To complete the machining, the eroded end of the electrode is removed, and after the necessary electrical adjustments have been made, the electrode is again fed directly into the workpiece. In appearance, the machine resembles a vertical drilling machines. It has a pedestal-type base, an electrode feed and control mechanism and a remote-unit power supply. The electrode feed is automatically controlled by an amplidyne and associated circuits to allow optimum cutting speeds to be maintained for a given cutting condition. With the machine work-table, provision is made for longitudinal, lateral and vertical traverse, and full rotation for work alignment.

A dielectric fluid, such as fuel oil, kerosine, or a special compound, Dielectro X, is used to enclose the cutting operation. There are two reasons for immersing the work in a dielectric fluid: first, to build up electric resistance so that the energy storage devices in the machine may be fully charged before discharge, and second, to flush the loosened particles from the work area. Fluid is lost only in negligible quantity through evaporation and carry-off on the work.

A highly conductive, easily machined material is used for the electrodes. Brass is commonly used, but other conducting materials may be employed. The electrode form is usually a negative of the shape to be produced. This allows the machining to be effected with the head moving vertically, When the machine has to revolve or the work-table traverse, for an irregular internal contour, an electrode of brass wire bent to the necessary working angle is all that is required, since the tool does not make physical contact with the work. (1990)

Catalogue Library

THE Engineering Centre, 351, Sauchiehall Street, Glasgow, C.2, has opened a Catalogue Library which contains the literature of more than 3,000 firms and also a complete set of British Standards. The facilities offered by this source of information are already being put to good use by consultants, designers, draughtsmen and buyers, as well as by many overseas visitors.

Manufacturers who are not already collaborating are invited to forward their current literature. (1988)

FATIGUE OF STEELS

Data on Fatigue and their Application to Design

This article is based mainly on the first part of the M.I.R.A. report entitled The Fatigue Strength of Steels *by R. J. Love, at the end of which is a list of 260 references covering almost every aspect of the subject. The second part of the report, dealing with the influence of material condition and surface treatment, has not been incorporated here as it is to be the basis of a paper to be contributed to a symposium, on the* Properties of Metallic Surfaces, *that will be held by the Institute of Metals in the Autumn of* 1952.

EVERY engineer understands how to use the well-known stress-strain curves for materials and how important are the ultimate failure and yield points to the designer. On the other hand, the application of fatigue data, such as S-N diagrams and Goodman diagrams, to design is not so widely appreciated. It is rather surprising that this state of affairs should exist in the motor industry, particularly as most of the failures that occur are due to fatigue. However, it is probably largely attributable to the fact that insufficient information is available as to the numbers of stress reversals to be expected during the lifetime of a vehicle, particularly in the case of the structure.

Research on this subject could furnish very useful information to the industry. The national research establishments test materials and supply basic data regarding their properties, such data being of general use to the whole of engineering industry. On the other hand, it would appear to be the duty of each individual industry to conduct the research necessary for the application of this data in their own narrower and specialized fields of operation. A certain amount of work has been carried out on this aspect of the problem by M.I.R.A. and other establishments, but there is still a great deal more to be done.

Before discussing how design criteria obtained as a result of research can be applied, it is necessary to describe briefly something of the nature of fatigue and the methods of presentation of data. Under a static load a material will, of course, fail at its ultimate tensile stress, but under fluctuating loads it may fail at a stress lower than that. Fewer cyclic variations are needed to produce failure at a high stress than at a low one, and there is a stress level below which the number of reversals to produce failure is so large that, to all practical intents and purposes, failure due to fatigue will not occur. This is termed the *limiting fatigue stress*, the *fatigue limit*, or the *endurance limit*, and it must be qualified by specifying the number of alternations necessary to produce failure at this stress.

Failures occurring as a result of many applications of stress may be easily recognized, as they usually show very little distortion even though the material may be, according to a tensile test, very ductile. The fractured section usually shows two separate features; a brightly polished zone in the area of initial rupture that is almost invariably at the surface, and a rough, crystalline zone in which the crack grew with increased rapidity until the final breakdown.

It is well known that the fatigue strength of a part may be reduced by the introduction of some surface discontinuity such as fillet, keyway, or hole. The amount of reduction of strength brought about in this way is a measure of what is called the *notch sensitivity* of the material. Laboratory measurements of notch sensitivity are usually made by comparing the fatigue strength of plain bars with that of bars carrying circumferential notches or transverse holes. A fact not so widely appreciated is that notch sensitivity is more pronounced in steels of high tensile strength, so that for notched parts the use of a material of this nature may not result in a proportionately high fatigue strength. On the other hand, sensitivity does not appear to be definitely related to impact strength, ductility or damping capacity.

Size is another important factor influencing fatigue strength. The limiting fatigue stress of a component of a particular shape is usually reduced as its size is increased. It is most marked in diameters up to about 1 in, but it does not alter appreciably with greater diameters.

S-N diagrams and their application to design

The fatigue characteristics of a material are usually shown on an *S-N diagram*, or *fatigue curve*, in which the number of cycles of stress N to produce failure is plotted against the stress S. Three methods of plotting the S-N diagram for an 80-ton steel are shown in Fig. 1. The relationship between S and N is perhaps best seen when N is plotted to a logarithmic scale. Frequently both S and N are plotted in this manner, but in most cases the difference between the semi-logarithmic and the logarithmic method of plotting is not very great. A useful feature of these two methods of plotting is that they produce a fatigue curve that often can be conveniently represented by two straight lines: a sloping line showing the increase of cycles for decrease of stress, joined by a horizontal line at the limiting fatigue stress.

For steels, the knee of the fatigue curve usually occurs between $0{\cdot}5\times10^6$ and $5{\cdot}0\times10^6$ cycles of stress. The values vary with the hardness of the steel, in general, the harder the steel the lower the value. Only a little extra knowledge is gained by continuing fatigue tests past $5{\cdot}0\times10^6$ cycles, Ref 1. However, the existence of an absolute fatigue limit within this range of cycles does not appear to be likely, as failures occurring at tens of millions of cycles have occasionally been encountered, Ref 2.

A useful estimate of the limiting bending fatigue stress for a steel may be obtained from the ultimate tensile strength. The ratio:

$$\frac{\text{rotating beam limiting fatigue stress}}{\text{ultimate tensile stress}}$$

is called the *endurance ratio*, and for a large number of steels it is about 0·5. Only a few have an endurance ratio outside the range of 0·4-0·6. An estimate of the limiting fatigue stress can also be made from a hardness measurement, since the Brinell hardness number is related to the ultimate tensile strength. B.H.N. $\times$ 0·1 will give an approximate value in tons/in^2 for the limiting stress, Ref 1.

There are, of course, other types of fatigue failure besides those induced by bending. Under axial loading the limiting stresses are usually lower than those obtained by the rotating beam method. In general, three explanations have been given that might account for the difference. Under axial load a greater volume of material is subjected to the maximum stress, so that there is a greater probability of there being a weak spot in the test specimen subjected to high stress. In the second place, it is very difficult to obtain true axial loading and, as a result, bending stresses are probably superimposed on the axial stress. Furthermore, stress gradient is important in relation to

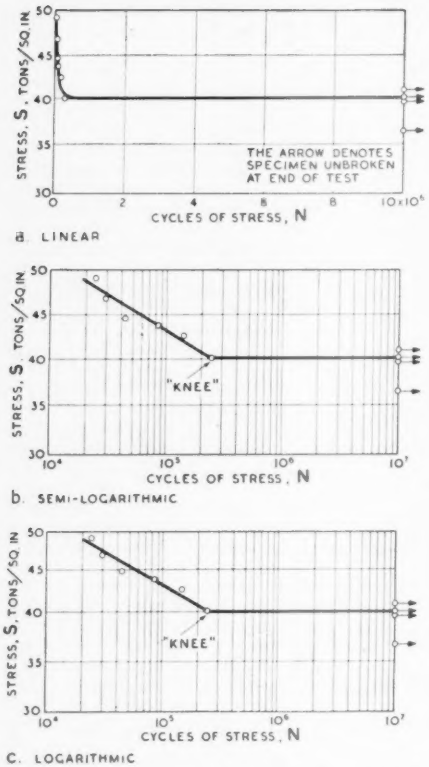


Fig. 1. An S-N diagram for a high carbon, 80-ton steel (rotating beam tests)

fatigue: material under a low stress gradient cannot withstand such a high stress as material under a high one. The average ratio:

$$\frac{\text{axial loading fatigue strength}}{\text{rotating beam fatigue strength}}$$

for tests on 42 steels was 0·82, and of these, 90 per cent were within the range 0·60-1·00, Refs 3, 4, 5 and 6.

Torsional fatigue tests usually give limiting stresses lower than the rotating beam values. For 48 steels, Ref 1, the average value of the ratio:

$$\frac{\text{limiting shear stress}}{\text{limiting rotating beam stress}}$$

was 0·56, the ratios being within the range of 0·45-0·65 for 90 per cent of the results.

As far as design is concerned the limiting fatigue stress is used for components that must withstand certain loads for a very large number of alternations. If the component is to have a definite life, or is only likely to undergo a limited number of reversals, a higher stress as indicated by the S-N curve may be used.

Goodman diagrams and the influence of stress range

So far, fatigue properties have been considered only under conditions in which the stress alternates between tensile and compressive values of equal magnitude. In practice, however, most components are subjected to cycles of stress whose upper and lower values are of the same sign, that is to say, either tension or compression; or between limits of tensile and compressive stress that are not equal in magnitude. Under bending and axial loading, the limiting stress range usually becomes smaller as the mean stress of the range is increased towards the ultimate stress value.

This tendency is best illustrated by means of a *Goodman diagram*, Fig 2, in which a line CB is drawn through the origin at 45 deg to represent the minimum stresses applied. The point B represents the static ultimate stress for the material, and AC is drawn to represent the limiting range for fully reversed stresses, that is, when the mean stress is zero. Then AB represents the limiting maximum stress that may be applied, and DB is the mean stress line. For example, when the mean stress has a value represented by E, the maximum and minimum stresses of the limiting range are represented by F and G respectively. Goodman proposed that the limiting fatigue stress or, in other words, half the limiting stress range when the mean stress is zero, should be taken as one-third of the static ultimate stress. An alternative is a *modified Goodman diagram* in which AC is drawn to represent the limiting stress range as found by experiment.

Influence of mean stress on limiting stress

Illustrated in Fig 3 are diagrams similar to those given by Smith in Ref 7. They are based mostly on the results of tests on axially loaded specimens, although some bending test results are included since, in this case, failure also occurs as a result of normal stresses, as opposed to shear stresses. In the diagrams on the right, half the stress range, or the limiting alternating stress, is expressed as a fraction of the fully-reversed limiting alternating stress.

An approximation to the variation of limiting range of stress with mean stress is given by line AB which, for most steels, will give conservative values for the limiting range of stress. This line does, in fact, agree with the modified Goodman relationship. The shaded area of the diagram is that in which failures occurred. A study of the shape and disposition of the area of failures suggests that in a few cases, in the range of low mean stresses, the alternating stress range is only slightly influenced by the mean stress. Furthermore, in these cases the alternating range is reduced in the range of high means stresses for the sole reason that the maximum stress of the cycle reaches the ultimate stress.

When the mean stresses are in the higher compressive range, failures would appear to be indirectly due to compressive yielding rather than to direct fatigue breakdown. For instance, attempts have been made to apply cycles of stress that are completely compressive by subjecting a T-beam to repeated loading from zero to a maximum value in such a way that high compressive loads were induced in the web of the beam. Failures were obtained in the part of the cross-section nominally loaded in compression, but it was believed that yielding occurred in the first few cycles so that, upon the release of the compressive load, residual tensile stresses were induced. The actual stress cycle imposed was thus one consisting of a high compressive stress followed by a lower tensile one.

Referring to the left-hand diagram of Fig 3a, illustrating the effect of mean stresses in the compressive range, the shaded area is again that in which failures have occurred. From the horizontal disposition of the straight line, it follows that the limiting stress range, in general, remains constant until higher mean stresses are reached. The maximum compressive stress is expressed as a fraction of the compressive yield strength which, in compression, is the criterion rather than the ultimate strength. The compressive yield strength is approximately equal to the tensile yield strength which varies from 0·5 of the ultimate tensile

strength in the case of annealed mild steel to 0·8 for heat-treated high tensile alloy steels, Ref 8. It should be realized that the mean stress changes from compression to tension only when an appreciable value of the ratio :

$$\frac{\text{maximum compressive stress}}{\text{compressive strength}}$$

has been reached. Ranges of stress in which both the minimum and the maximum values are compressive only occur at the extreme left of Fig 3a.

For notched specimens, Fig 3b, the relationship between the limiting alternating stress and the mean stress, for tensile mean stresses, is similar to that illustrated in Fig 3a. It must be remembered, however, that the reversed stresses for notched specimens are considerably less than for unnotched ones of the same material. From Fig 3b it appears that the limiting alternating stress for most steels in the notched condition may increase considerably for the higher compressive mean stresses.

In torsion, it can be seen from Fig 4a that the limiting shear stress remains approximately constant so long as the maximum stress of the cycle is below about 0·8 of the shearing yield strength. On the other hand, Fig 4b shows that for notched torsional specimens the limiting alternating stress is reduced as the mean shear stress is increased. This variation can be approximated by a Goodman type of relationship. It should be noted that in this case the mean shear stress is expressed as a function of the torsional modulus of rupture. Smith points out that for 35 steels the ratio :

$$\frac{\text{torsional modulus of rupture}}{\text{tensile strength}}$$

varied from 0·68 to 0·96 with an average value of 0·83, the higher ratios being obtained with the softer steels.

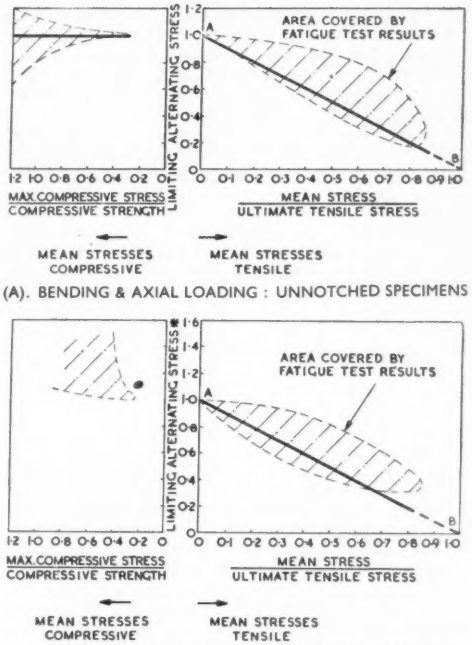

AREA COVERED BY FATIGUE TEST RESULTS
LIMITING ALTERNATING STRESS
MAX. COMPRESSIVE STRESS / COMPRESSIVE STRENGTH
MEAN STRESS / ULTIMATE TENSILE STRESS
MEAN STRESSES COMPRESSIVE
MEAN STRESSES TENSILE
(A). BENDING & AXIAL LOADING : UNNOTCHED SPECIMENS
(B). BENDING & AXIAL LOADING : NOTCHED SPECIMENS

Fig. 3. Influence of mean normal stress on limiting stress range

Criteria of failure under combined stresses

Of the many criteria of failure that have been examined in connection with fatigue, the distortion energy or total shear energy theory of Mises-Hencky appears to give the best results. Any tri-axial system of normal and shear stresses can be reduced to three normal principal stresses acting in mutually perpendicular planes, and for principal stresses P_1, P_2, and P_3, the distortion energy, D, is given by:

$$D=\frac{1+\mu}{3E}\left[P_1^2+P_2^2+P_3^2-(P_1P_2+P_2P_3+P_3P_1)\right]$$

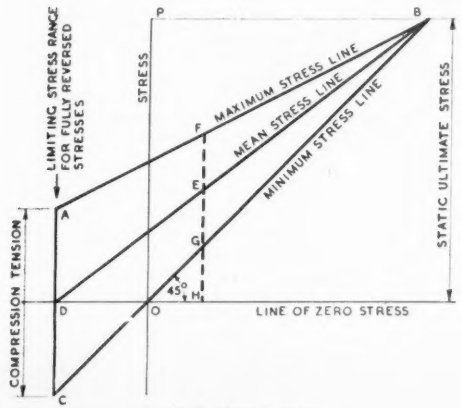


Fig. 2. Goodman diagram

where E = modulus of elasticity
μ = Poisson's ratio.

In most fatigue problems only surface stresses are important so that $P_3=0$, and the expression becomes :—

$$D=\frac{1+\mu}{3E}(P_1^2+P_2^2-P_1P_2)$$

When considering the distortion energy theory as a criterion of failure for a specified material, the expression is used in the form:

$$P_1^2+P_2^2-P_1P_2=\text{constant}$$

and the constant can be determined from simple fatigue tests on the material.

Knowing the rotating beam strength of the material it is possible, by carring out stress measurements on a part, to estimate from the criterion of failure its reversed load fatigue limit, Ref 9.

Marin, Ref 10, claims responsibility for the development of the distortion energy theory for fatigue problems. He considers the case of combined stresses in which the mean of each of the applied stresses may have any value, but in which the stresses are applied at the same frequency and reach their peak values at the same time. He suggests that failure will occur when the distortion energy, at the peak applied stresses, is equal to the limiting value under a uni-axial fluctuating stress of the same mean value as that given by the mean combined stresses. In other words, it should be possible to reduce the state of fluctuating combined stress to an equivalent fluctuating uni-axial stress that can be examined in relation to the previously described "mean stress-limiting stress range" diagram.

It appears that this distortion energy criterion gives compatible results for the special case of combined stresses with zero mean stresses, and for a few results obtained for repeated stress. It does not, on the other hand, appear to

give good results for the few tests that have been done with a more general type of loading. It should be possible, using the appropriate failure criterion, to predict the shape if the "limiting alternating shear stress—mean shear stress" diagram from the "limiting alternating normal stress—mean normal stress" diagram. However, the distortion energy criterion used in this way does not appear to give favourable results. It is clearly desirable that more tests be carried out to investigate fatigue strength under combined stress.

Gough and Pollard, Refs 11 and 12, carried out tests in which reversed normal stresses were imposed at the same time as reversed shear stress, both stresses reaching their zero values at the same time. Their tests gave results that were fairly consistent with the distortion energy criterion, but they suggest that the best method of determining the values of a normal stress f and a corresponding shear stress q to produce failure is to use the ellipse equation :

$$\frac{f^2}{F^2}+\frac{q^2}{Q^2}=1$$

where F = fatigue limit under simple bending
Q = fatigue limit under pure shear.

For specimens containing a sharp V notch they found that the ellipse equation would give approximate values, but better results could be obtained with another more complex, equation that they present. Later, Gough showed that this applied also to specimens with a transverse hole, and to specimens with a sharp fillet. He further demonstrated that splined specimens represented a special case, the ellipse equation applying when the bending stresses were greatest, and the other one applying where the torsional stresses were greatest. The same would appear to be the case for Puchner's tests on specimens with a keyway.

In a test on one steel only, it was shown that both the ellipse equation and the other one could be used for unnotched and notched specimens respectively when a static bending stress is superimposed on the reversed bending stress, and a static torsional stress is superimposed on a reversed torsional stress. However, in these cases the values of F and Q are the values for the limiting reversed bending and torsional stresses, obtained in the presence of the static stresses.

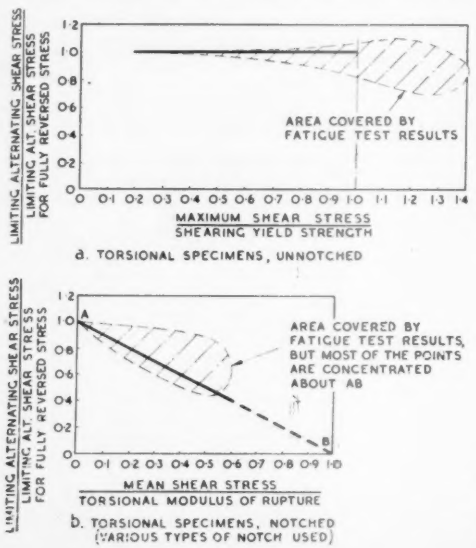

LIMITING ALTERNATING SHEAR STRESS
LIMITING ALT. SHEAR STRESS FOR FULLY REVERSED STRESS
AREA COVERED BY FATIGUE TEST RESULTS
MAXIMUM SHEAR STRESS / SHEARING YIELD STRENGTH
a. TORSIONAL SPECIMENS, UNNOTCHED
AREA COVERED BY FATIGUE TEST RESULTS, BUT MOST OF THE POINTS ARE CONCENTRATED ABOUT AB
MEAN SHEAR STRESS / TORSIONAL MODULUS OF RUPTURE
b. TORSIONAL SPECIMENS, NOTCHED (VARIOUS TYPES OF NOTCH USED)

Fig. 4. Influence of mean shear stress on limiting shear stress range

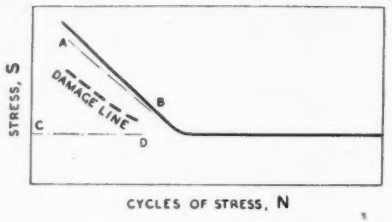


Fig. 5. The damage line

Under-stressing, over-stressing and impact

The term *under-stressing* refers to the action of subjecting a part to cycles of stress below its normal fatigue limit before testing it at a higher stress. With many steels this treatment improves the strength of the part considerably. For example, by subjecting a part to a stress below the normal fatigue limit and then raising the stress in small steps, applying each stress level for a large number of cycles, increases of fatigue limit of about 20 per cent have often been obtained. An increase of static tensile strength due to under-stressing has also been observed, Ref 13.

The term *over-stressing* refers to the loading of a part above its normal fatigue limit for a number of cycles before loading it at a lower stress. When an over-stress has been applied for a relatively short time, an improvement of properties, similar to that found with under-stressing, has been found. However, if the over-stress is very high, or if a low over-stress is applied for a large number of cycles, the part then becomes damaged, by which it is meant that the fatigue strength is lowered. The stress and corresponding numbers of stress cycles that may be imposed without causing damage, as measured by the effect on the fatigue limit, for example, may be indicated on the normal S-N diagram by a damage line, as in Fig 5. The damage line may vary for different steels roughly within the limits indicated by the lines AB and CD.

Most components are subjected in service to fatigue loads of a continuously varying intensity, but only a small amount of test experience has been obtained using loads of this kind. Muller-Stock, Gerrold and Schulz, Ref 15, showed that by alternately over-stressing and under-stressing, damage may not be done until an abnormally large number of cycles have been applied. This was also found in many of the tests carried out by Dolan, Richart, and Work, Ref 16, and among their many interesting observations were the following :

(a) In general, repetitions of an under-stress alternated with repetitions of a stress 10-20 per cent above the original fatigue limit, resulted in a marked increase of fatigue life.

(b) Cycles of under-stress alternated with cycles of stress considerably above the normal fatigue limit produced fatigue lives at the upper stress, which were the same as, or in some cases very much less than, those indicated by the normal S-N curve.

As the speed of fatigue loading is increased the condition of repeated impact loading is approached. Repeated impact tests give results similar in character to those of the usual S-N fatigue curve. McAdam, Ref 16, shows that the ratio of impact endurance limit to the rotating beam endurance limit for several steels is approximately constant. From this, the life to be expected for a steel subjected to impacts of comparatively low energy can be related to its

normal fatigue strength. On the other hand, for blows of high energy, the normal single blow impact strength gives a better indication of its behaviour.

Chafing fatigue

When slight oscillatory slipping occurs between two surfaces in contact fretting corrosion takes place, a cocoa-coloured deposit, mainly iron oxide, being formed. Fatigue failures that are promoted by this type of rubbing action are termed *chafing fatigue failures*, Refs 17-25. It is difficult to assess the amount of reduction of strength due to fretting corrosion alone, since at a clamp or press fit, a stress concentration effect exists. Chafing tests carried out using a clamping action have shown :

(a) There is a decrease of strength accompanying an increase of clamp pressure, although the actual amount of fretting corrosion may decrease with increase of pressure.

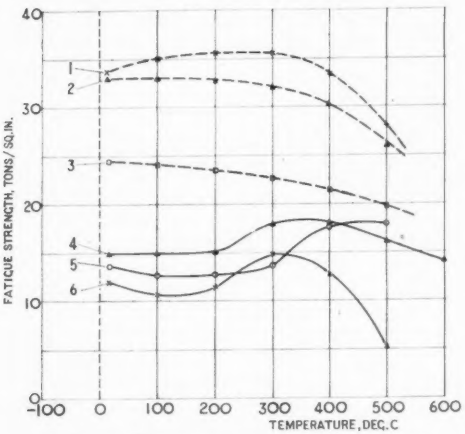
(b) There is a decrease of strength accompanied by an increase of hardness of the clamping material.

(c) Strength is not altered appreciably by using a specimen of considerably higher tensile strength. For a selection of steels, the strength has been found to be of the order of 13 tons/in^2.

(d) For some carbon steels and some austenitic stainless steels, better fatigue strength has been obtained in the annealed condition than in the cold-worked condition.

Warlow-Davies attempted to determine the deleterious effect of fretting alone by first producing a fretted surface on a rotating beam specimen, and then determining the strength of the specimen with the clamp removed. The strength of a medium carbon steel and that of an alloy steel were decreased in this way by amounts of the order of 10-20 per cent.

Fretting corrosion is most marked with highly finished surfaces. Furthermore, surfaces of like materials in combination are more susceptible than those of unlike materials. Stainless steel in any combination is most susceptible. Fretting corrosion may be reduced, but at the same time the fatigue strength may also be reduced in some cases, by :



1. 3.5 per cent Ni steel tempered at 425 deg. C—Moore and Jasper
2. 3.5 per cent Ni steel tempered at 650 deg. C—Moore and Jasper
3. 19 per cent Ni, 8 per cent Cr steel—Moore and Jasper
4. 0.14 per cent C steel—Lea
5. 0.06 per cent C steel—Lea
6. 0.02 per cent C Armco iron—Tapsell and Clenshaw
— — — —rotating beam tests
——axial load tests

Fig. 6. Effect of temperature on fatigue strength

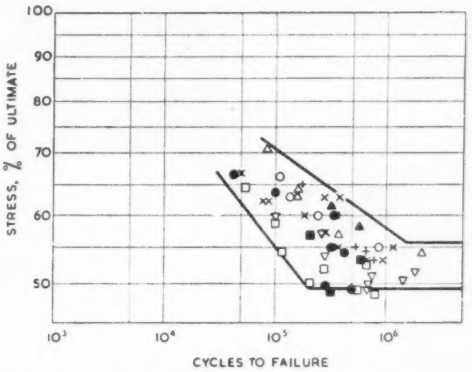


Fig. 7. The scatter of results illustrates the futility of relying on one or two tests only

(a) Nickel-plating.
(b) Chromium-plating.
(c) Using special lubricants that combine low viscosity with the property of maintaining a continuous film.
(d) Interposition of a soft material such as a thin coating of aluminium paint.
(e) Roughening the surfaces.

Increased fatigue strength under chafing conditions may result from such treatments as shotpeening, surface rolling, or carburising. Where a clamping action is present, the use of nitriding is very beneficial. With this treatment, increases in strength of from 70-100 per cent, under clamping loads, have been obtained.

Environment

Although the corrosion of specimens prior to testing produces a considerable reduction in fatigue strength, an even greater effect is evident if the specimen is submerged in the corrosive medium while being tested. Fatigue failures occurring in the presence of corrosive media are termed *corrosion fatigue failures*. The S-N curve obtained under these conditions is different from the normal one in that there is no indication of a fatigue limit: the curve continues to fall at a considerable rate even after a great number of cycles. Since the magnitude of the corrosive attack is dependent on the time of the exposure to the corrosive medium, rather than on the number of cycles of load applied, there is a pronounced speed effect. Thus, a specimen subjected to a low speed of loading will fail in fewer cycles than a specimen loaded at a higher rate. In the same manner, rest periods in the corrosive medium will reduce the total number of cycles to failure.

The loss of strength due to corrosion may be reduced by using stainless steels, or by using certain surface treatments. Zinc coatings are very effective, and so is enamel over an aluminium coating. Nitriding has been shown to be good, and a special silicon-impregnated case has been used effectively.

Some results of tests showing the effect of temperature on rotating-beam and axial-loading fatigue strength are given in Fig 6. It will be seen that with most of the steels the fatigue strength decreases slightly as the temperature is raised from room temperature to about 200 deg C. Above this, however, several of the lower-strength steels show an increase of fatigue strength which rises to a maximum and then falls rapidly. It is known that the tensile strength of most steels similarly falls, and then rises to a maximum and falls again. At high temperatures the fatigue strength is often greater than the creep strength,

so in these cases creep may be more troublesome than fatigue if the speed of loading is low. On the other hand, as the temperature is reduced below freezing point fatigue strength increases appreciably.

Application of fatigue data to design

All this information about fatigue has been obtained, of course, with the object of applying it to design, to increase efficiency by economizing in materials, saving weight, and obviating breakdown of machinery and structures. The practical application of this data is, therefore, an important aspect that has not, perhaps, received enough attention in the past.

Some work has already been carried out, mainly on gears, to determine the number of stress reversals that different components have to withstand. The Detroit gear Division of the Borg-Warner Corporation, Ref 27, have used the following figures for the full-load life of the gears in a three-speed layout: 140 miles in first gear and 420 miles in second. Then, using the axle ratio and tyre sizes, they convert the miles of life into a number of stress cycles. The reversals experienced by the constant mesh gears will, of course, be the sum of those for the other two. Almen and Straub, of General Motors Corporation, in Ref 28 give a full-throttle life of 100,000 cycles for the low ratio and 300,000 cycles for second gear, giving a total of 400,000 cycles for the constant-mesh pair. They also refer to formulae for making the stress calculations.

In design it has been the practice in most cases to devote reasonable care to the elimination of stress raisers in the engine and transmission, but in this respect the structure has been sadly neglected. Nearly all the manufacturers carry out extensive fatigue tests on prototype vehicles and on components, and then the design of parts that fail is modified to eliminate the cause of failures. If better use were made of existing data, and if research were undertaken to enable it to be applied more generally, a reduction would be possible of the amount of re-design needed at the development stage and, furthermore, greater reliability of the product would result.

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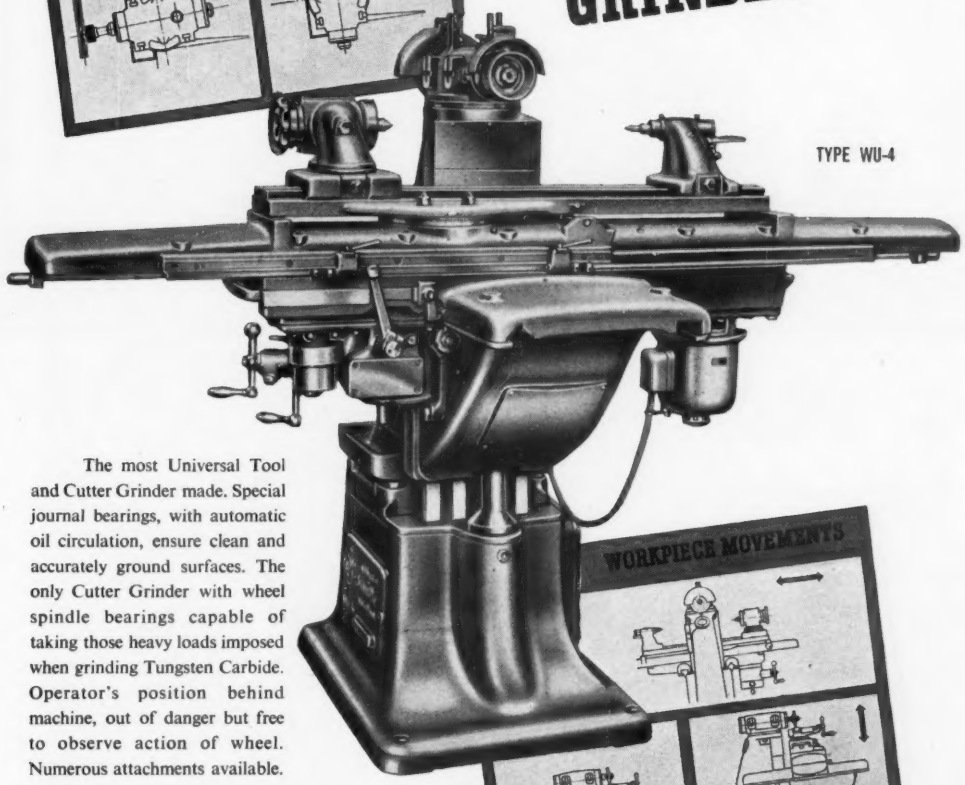

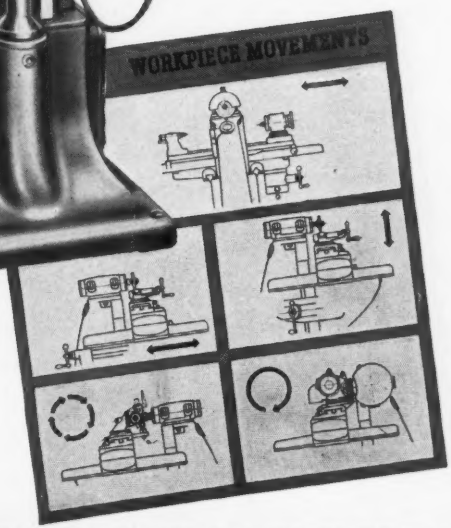

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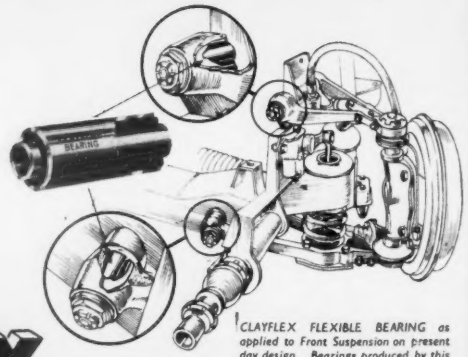

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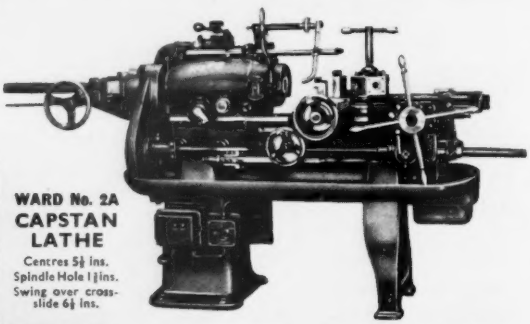

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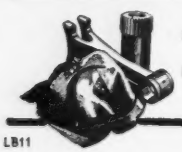

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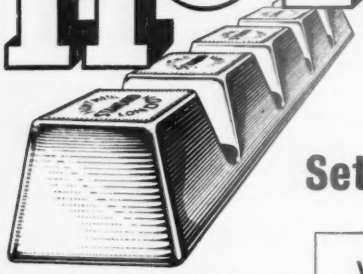

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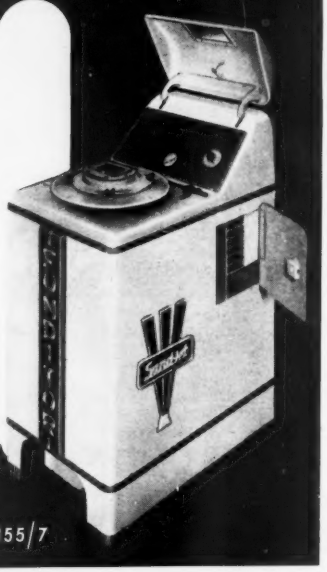

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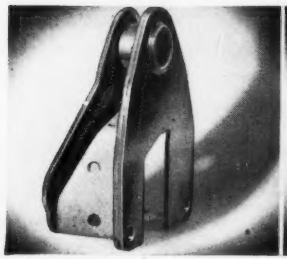

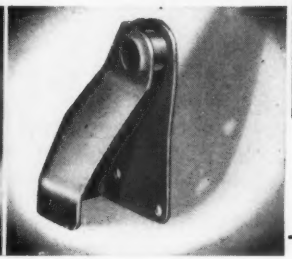

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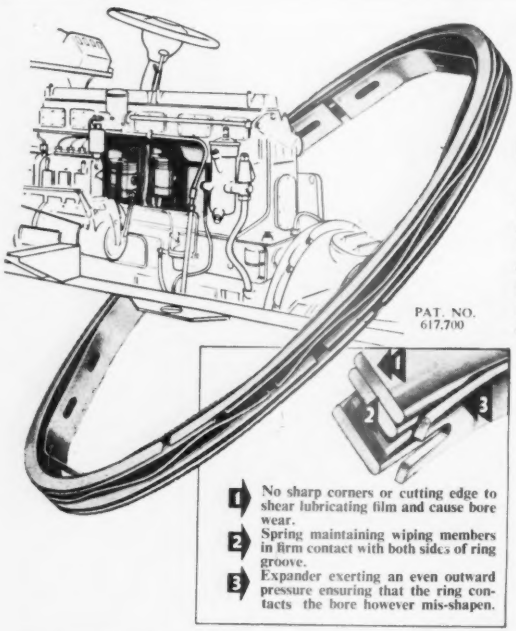

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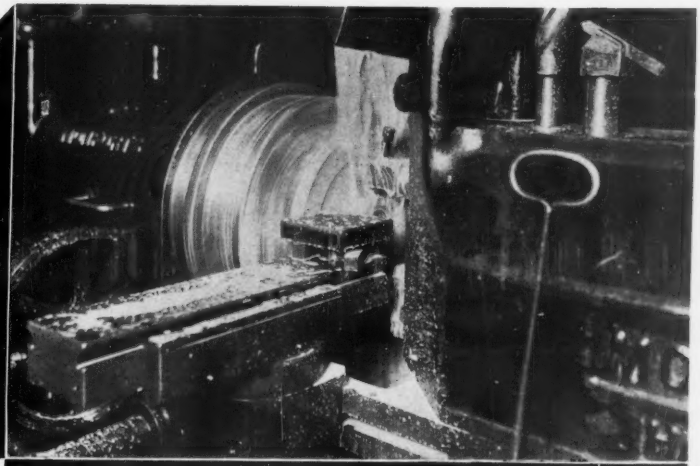

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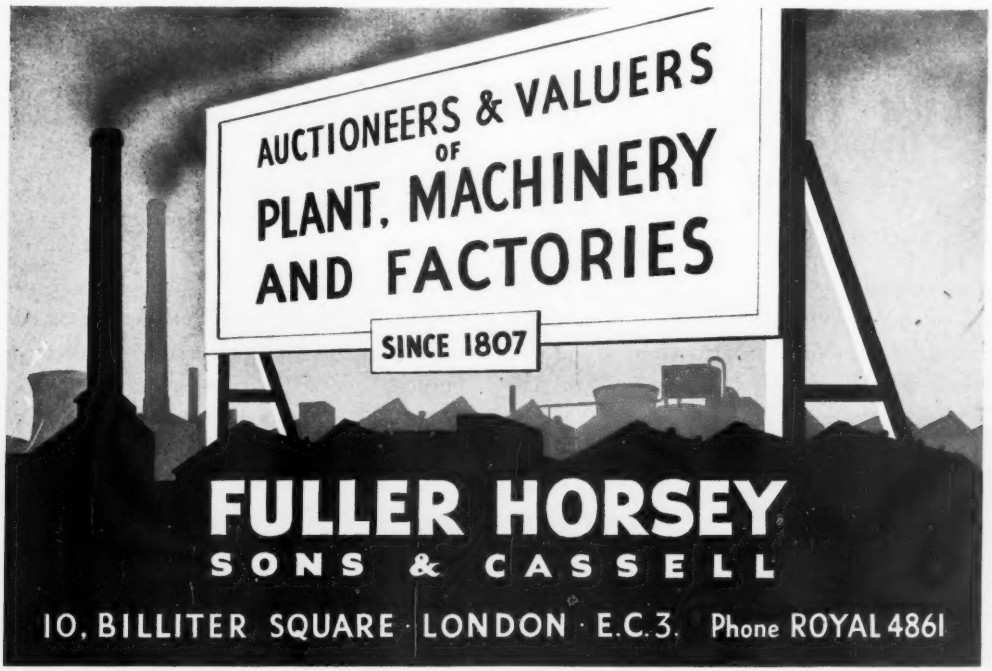

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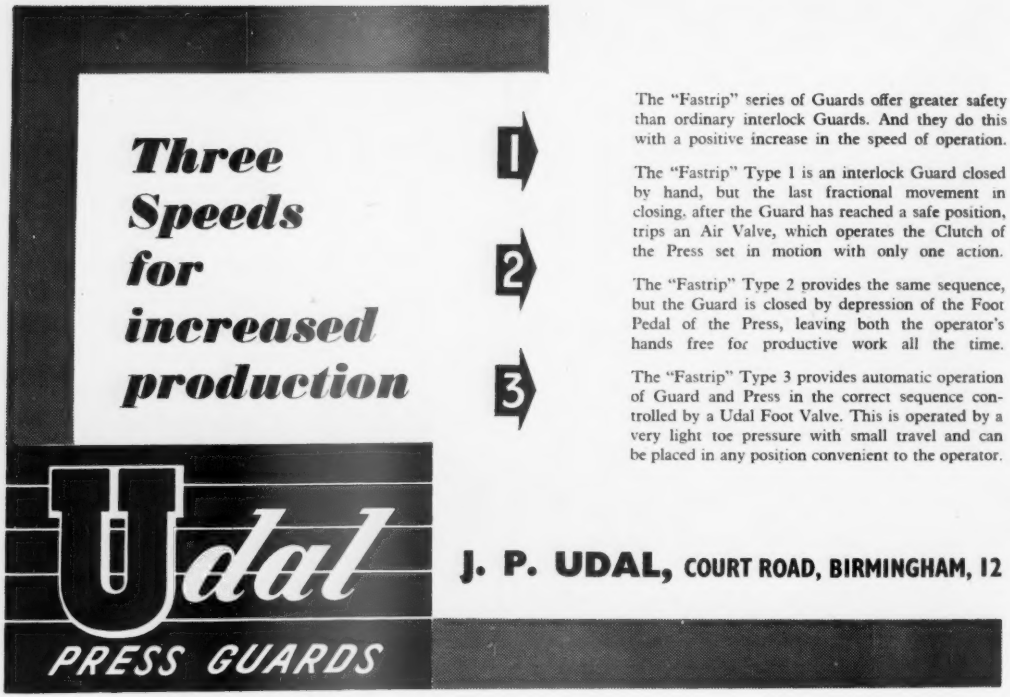
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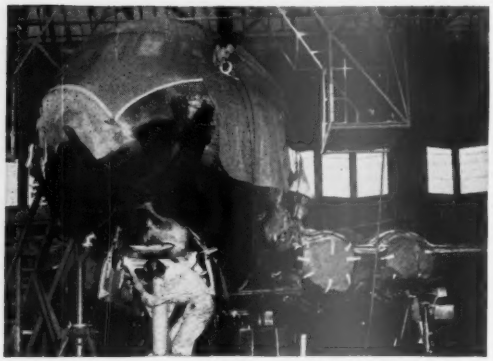

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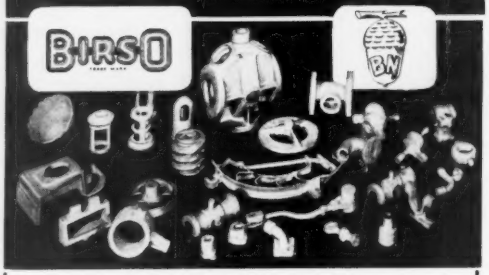

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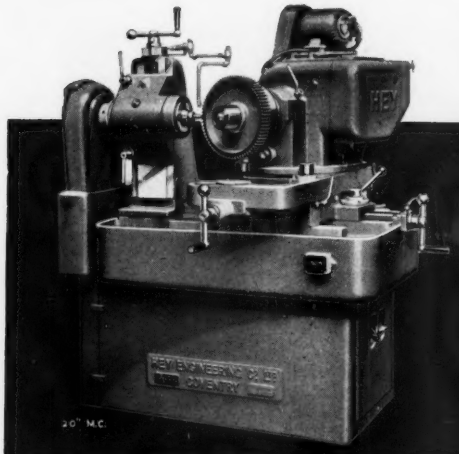

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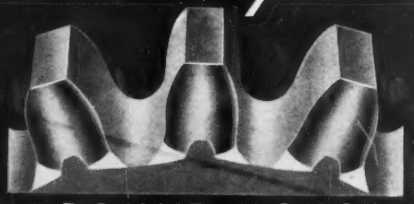

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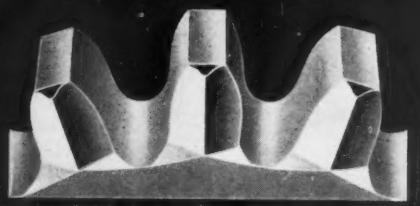

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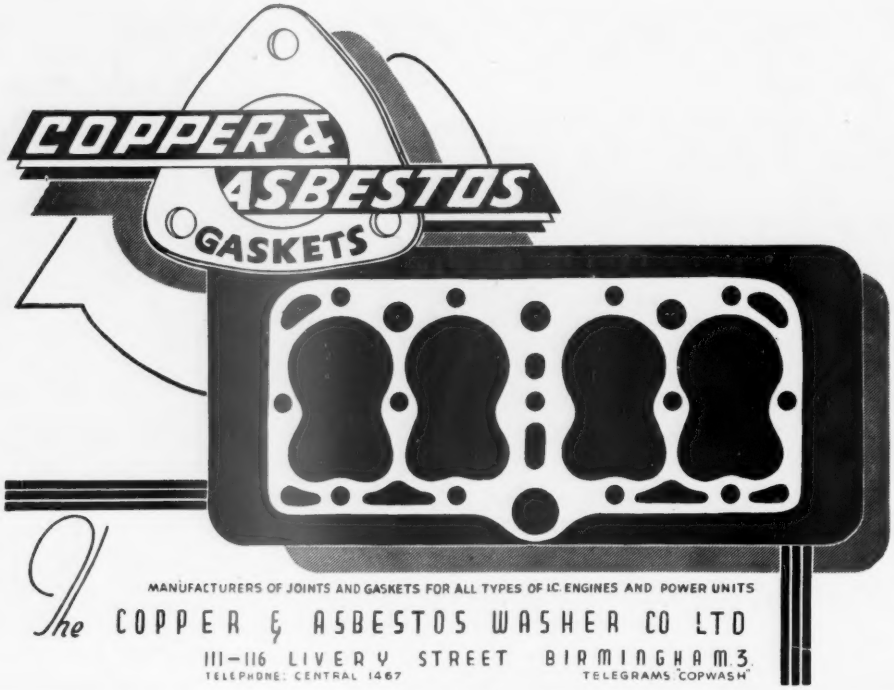

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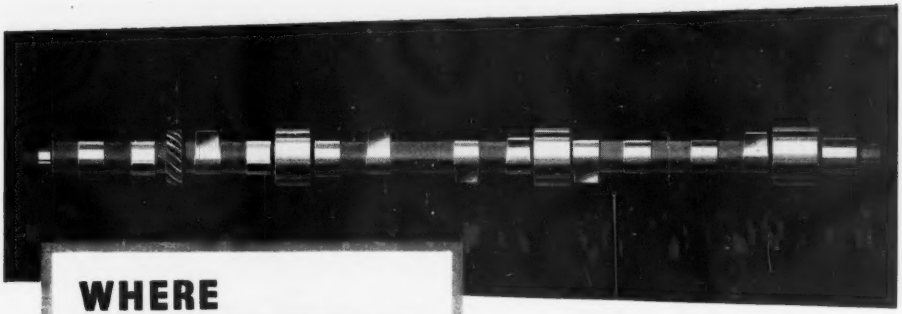

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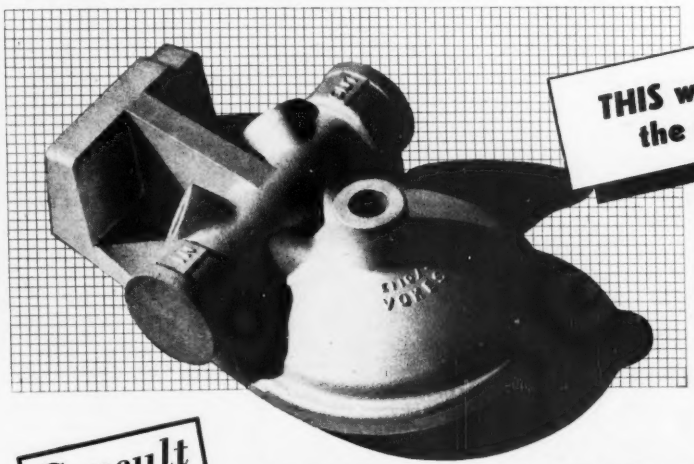

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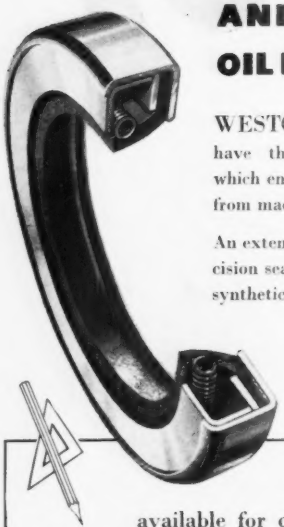

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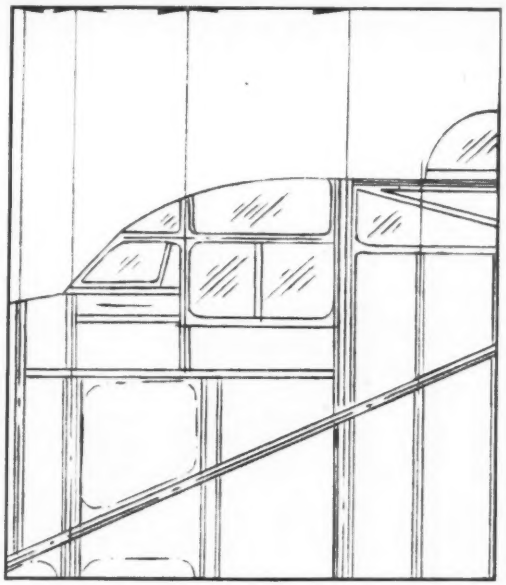

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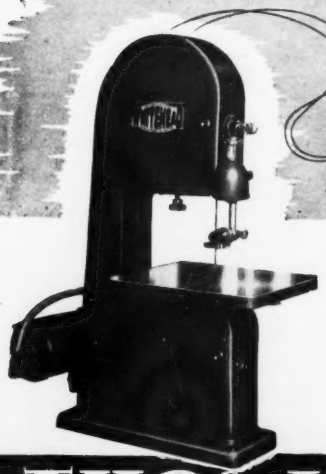

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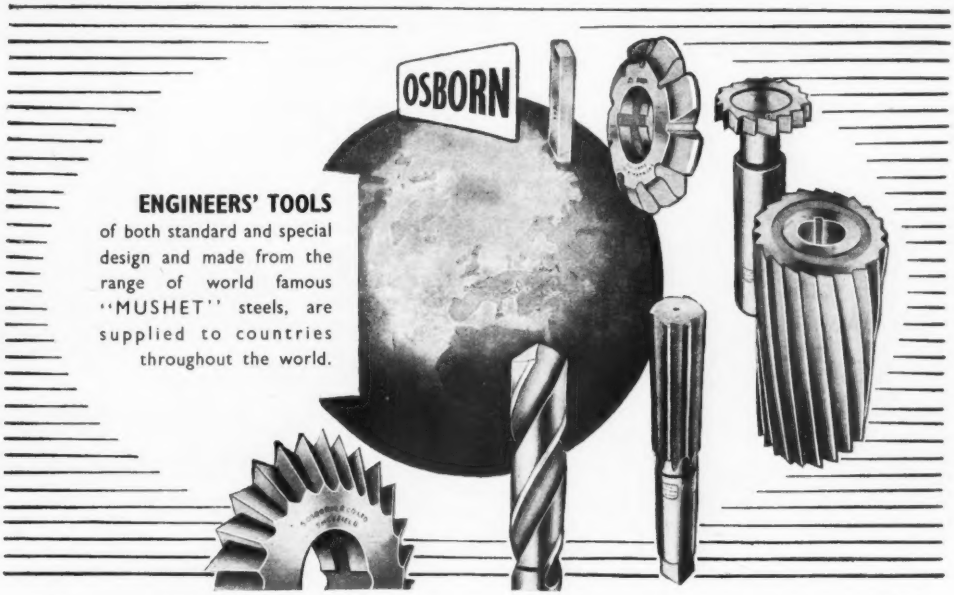

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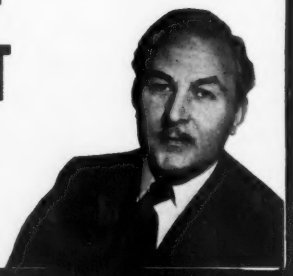

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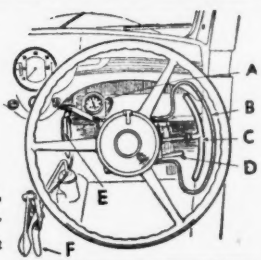

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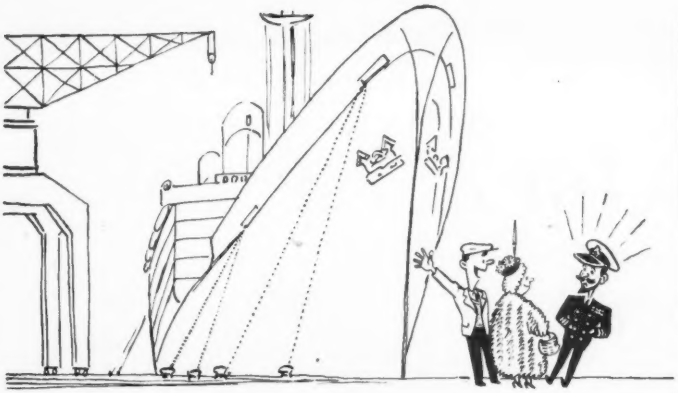

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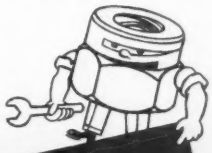

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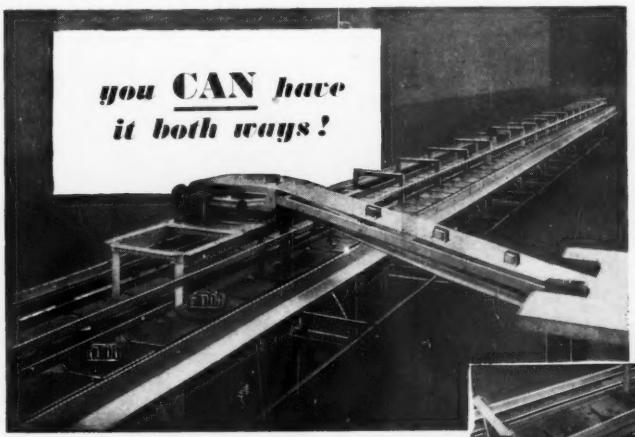

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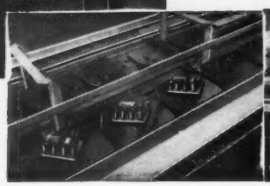

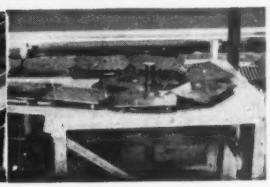

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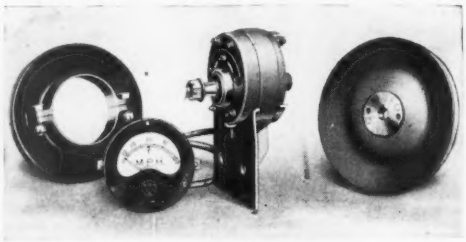

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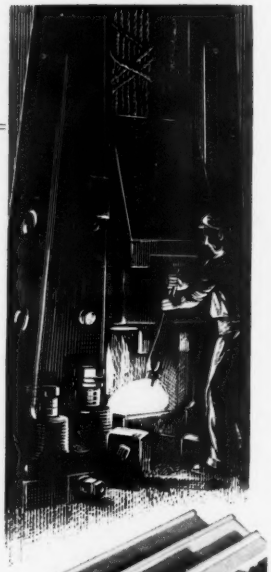

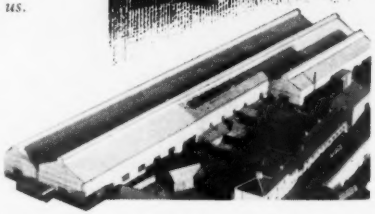

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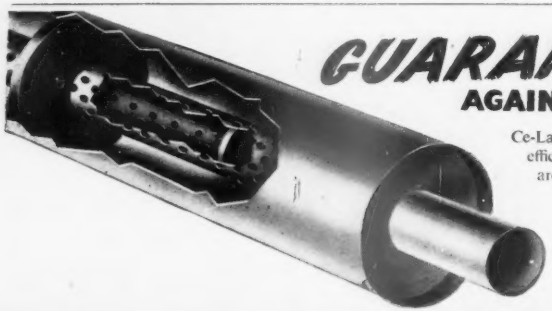

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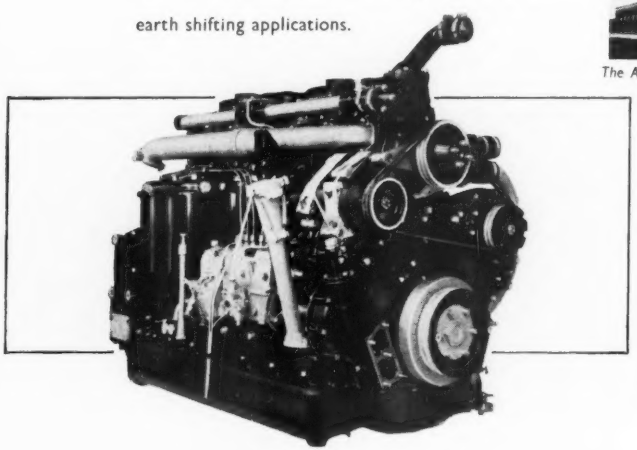

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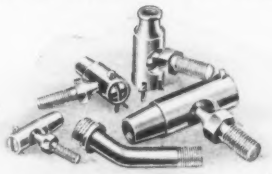

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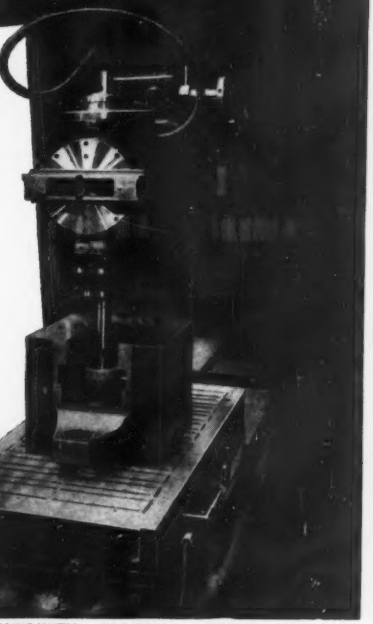

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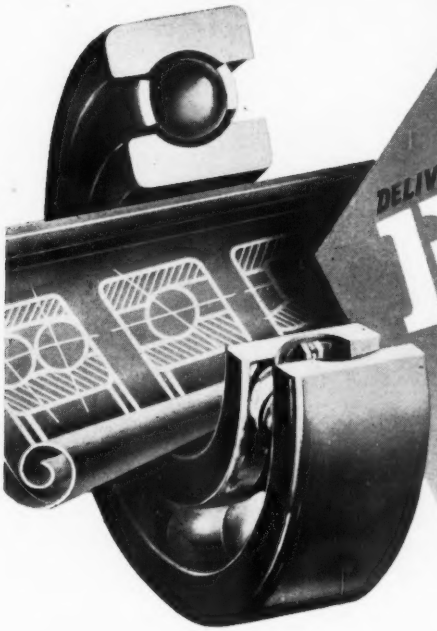

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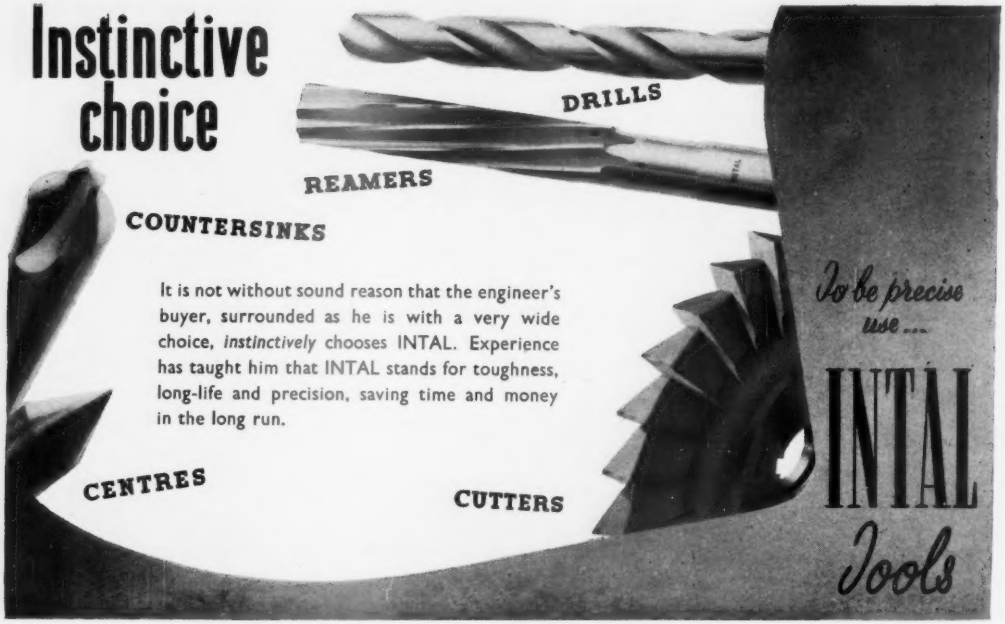
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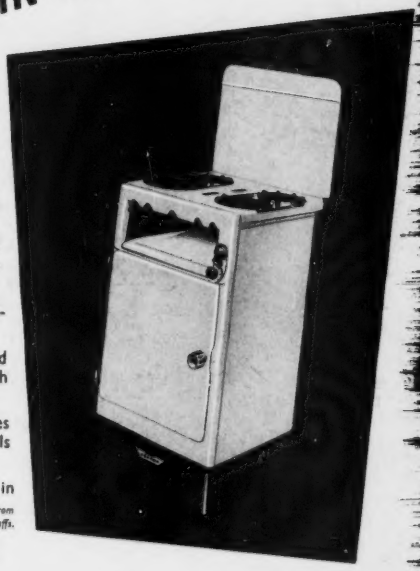

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